POWER

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Mental Indigestion

NOT long since, a young man who had had some experience around small steam plants applied to the chief engineer of a large mill in Southern New England for a position as oiler. His object was to get in touch with larger engines and their auxiliaries, hoping some day (so he said) to be able to take charge of a plant. He was hired. One day he asked the chief what books or mechanical publications he would advise him to take.

"Young man," said the chief, "you might take all or any of the books that are published and they would be of no benefit to you unless you read them and think about what you read. My advice to you is to get an engineers' catechism, and subscribe for the best paper published that is devoted to the interests of this profession, and read them and think."

The mere possession of any number of books and papers will not benefit a man or make an engineer of him, nor cause him to advance, no matter what they may contain. He must read them, digest them. Food taken into the stomach is of benefit to the system only when it is digested. Good seed sown on fertile soil will produce results, but the best seed sown on a gravel heap will produce nothing. The best medicine known to the medical profession will have no beneficial effect on the patient if allowed to remain in the bottle. Books and papers are of no value if allowed to remain unopened.

Ten years ago a drummer called in to see the chief engineer of a plant, and while waiting for the chief to make his appearance got into conversation with the oiler, who told the drummer that they had been using his firm's goods for eight years to his knowledge, for he had been oiling there that length of time and he knew.

Upon being asked what mechanical papers he read, he replied that he did not "go much" on mechanical papers, and when he got time to read he generally got hold of story magazines or something that was interesting as he had been in the business so long that reading mechanical papers was like reading a rehash of old chestnuts. That man is still an oiler after eighteen years.

A man who had run an engine in a good sized factory for over two years without having any serious mishap, made application for membership in an association of engineers. One of the investigating committee on his application called on him at his engine room, and among other things noticed several numbers of a technical journal still in the wrappers on top of a closet. This same man was asked in his examination, which end of a boiler feed pump should have the larger area of piston, and he replied that "he never noticed." On being asked what the object was in putting lap on a valve, he replied that he did not know that it had any lap.

Another applicant was asked in his examination to describe how the governor on his engine performed its function of regulating the speed and replied: "By the balls flying 'round." He knew they flew 'round, but there it ended. The same man frankly said he would send for a machinist to set his eccentric if it should ever happen to slip as he would not know how to set it right.

Surely in these days with books, papers, societies and schools within the means and easy access of any one possessing the desire to learn there is no excuse for such ignorance. Such men should either make good or make room. A man is apt to progress as far as he thinks.

The man who never noticed which end of his boiler feed pump had the largest piston certainly never overworked his "think tank" regarding the construction or operation of pumps, nor could the man who did not know his valve had any lap ever have done much thinking about using steam expansively.

Do not be satisfied just to do a thing; think why you do it. Ask yourself questions, mentally and try to find the answer. Do not be like a trained dog that jumps over a chair at the word of command, without knowing why he does it. If you are asked to open or shut a certain valve, do it, and then ask yourself why? What effect did it have when I closed that valve? Think out the answer.

You are a young man and there is no excuse for you to remain an oiler if you read and think.

[*Contributed by H. R. Low, Moosup, Conn.*]

Test of Lentz Engine

SYNOPSIS—A simple 15x21-in. engine with steam at 148 lb. pressure, superheated about 50 deg. and exhausting against atmospheric pressure, runs on 19.2 lb. of steam per indicated horsepower-hour.

When a third of a century ago the leading steam engineers of America, acting as the committee on tests at the Centennial Exposition, undertook to estimate the amount of steam required for a horsepower in order to apply that unit of capacity to the steam boiler, they decided upon 30 lb. per hour, and that was doubtless a very good performance for the four-valve noncondensing engine of that period. Recently engineers and engine builders have been talking about performances below 20 lb. for noncondensing engines, and about a half of that quantity for compound-condensing units with a high degree of superheat. It is interesting to have the authority of such engineers as Robert W. Hunt & Co., of Chicago, for such figures. We have had the privilege of reviewing a detailed report made by the engineers of that company of tests carried out by themselves at the Thorp Street plant of the People's Gas Light & Coke Co., of Chicago, upon Lentz engines built by the Erie City Iron Works.

These engines were installed some two years ago, and are directly connected to blowers used for handling the gas. Front and rear views of the installation, comprising both single-cylinder and compound engines, are reproduced in Figs. 1 and 3 respectively.

On Apr. 24, 1911, engineers from the office of R. W. Hunt & Co. ran a three hours' test upon one of the

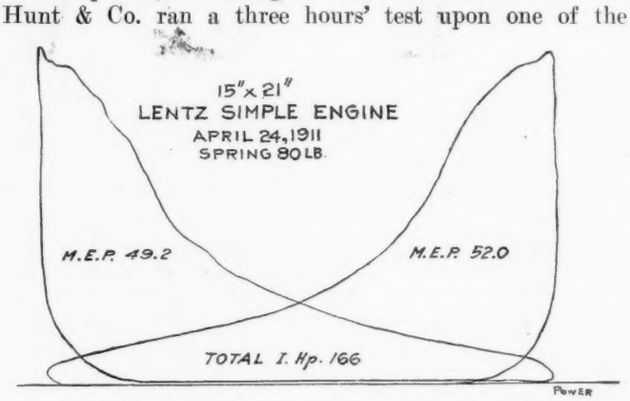

FIG. 2. DIAGRAMS FROM LENTZ ENGINE

simple engines. This developed an indicated horsepower with 19.2 lb. of steam per hp.-hr. The steam pressure at the engine ranged between 132 and 153 lb., averaging 147.6 lb. with an average superheat of 49.7 deg. F. The back pressure was that due to the atmosphere. The pro-

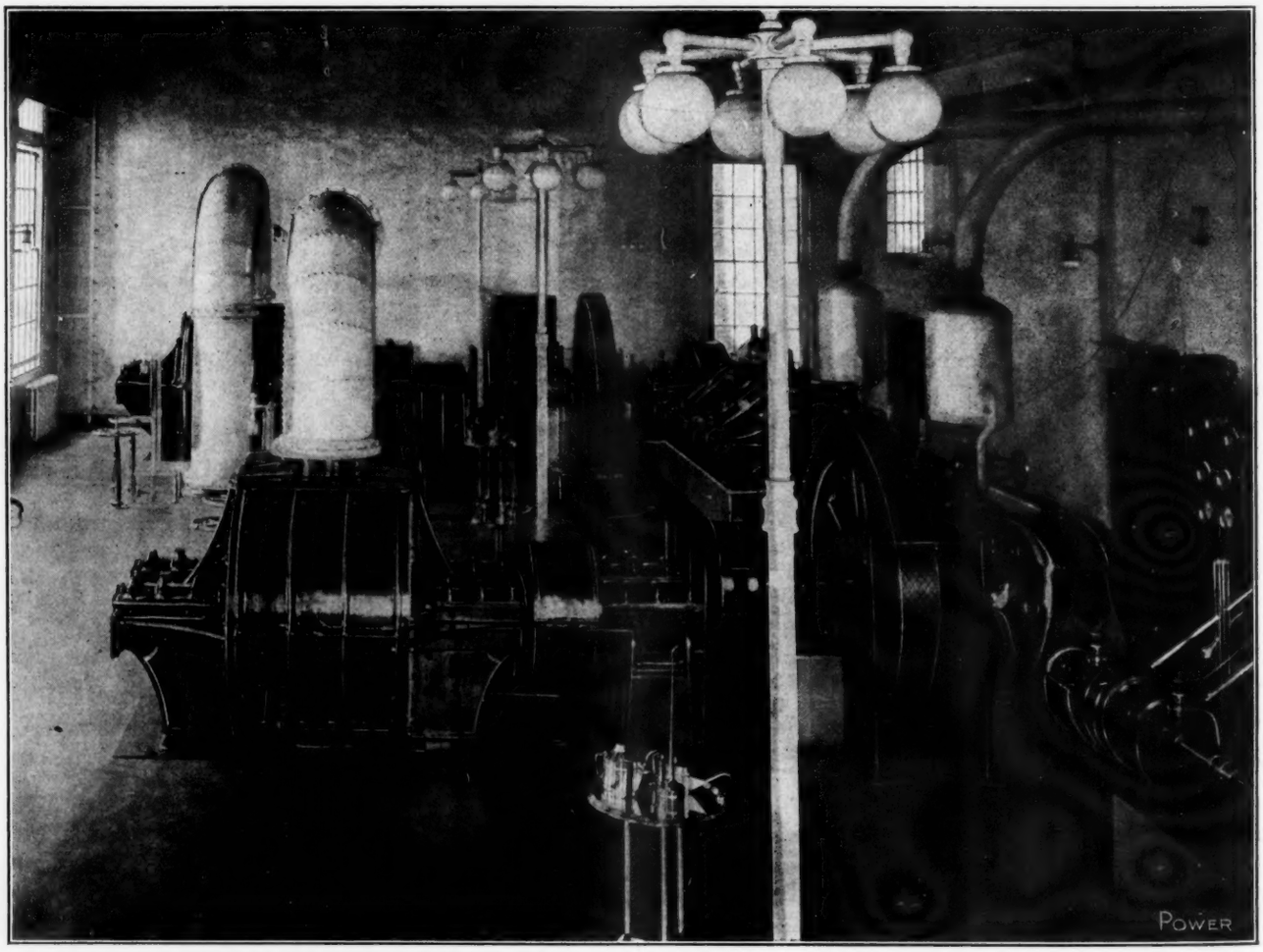

FIG. 1. FRONT VIEW OF LENTZ ENGINES IN THORP ST. STATION, PEOPLE'S GAS LIGHT & COKE CO., CHICAGO

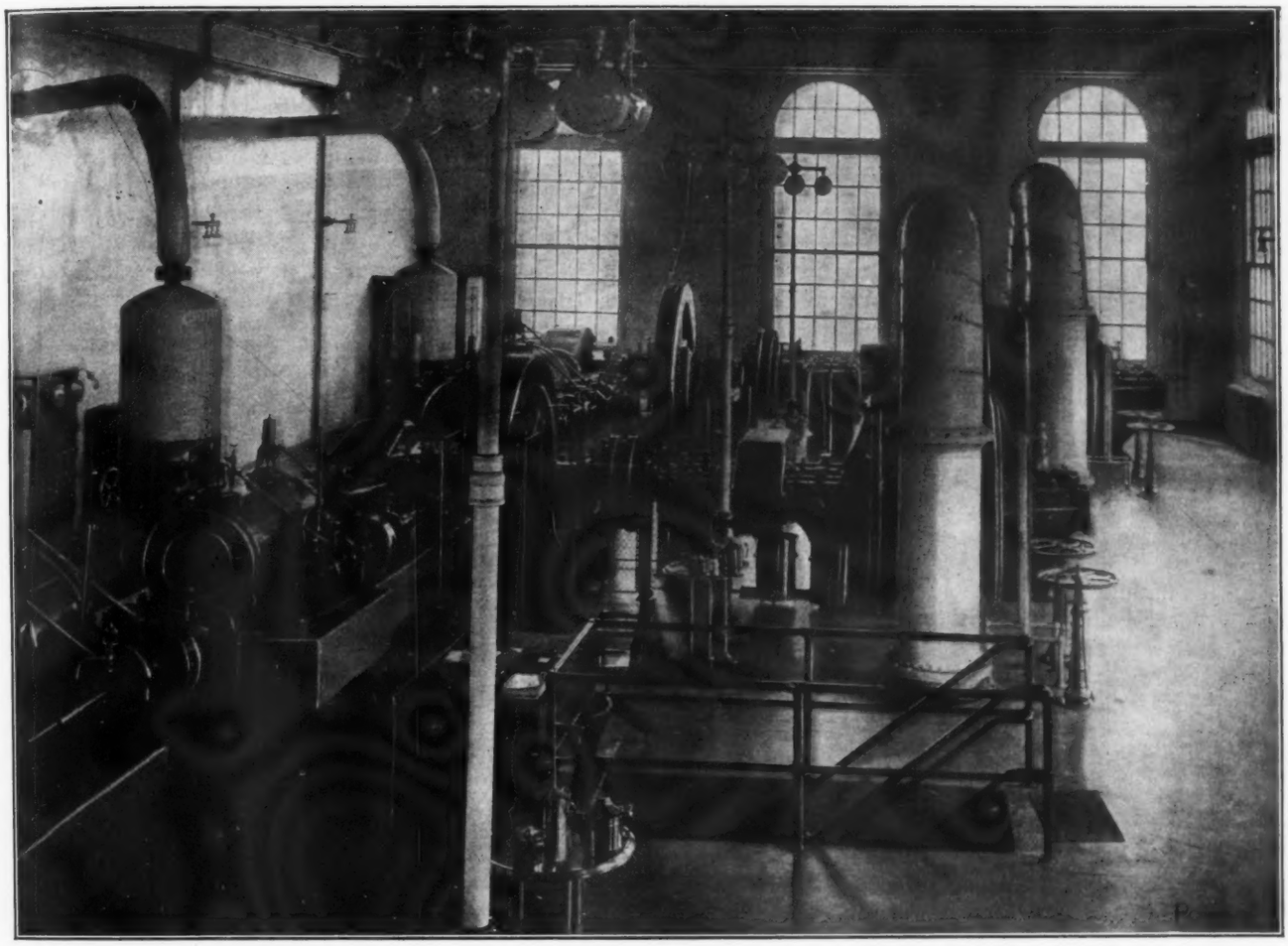

FIG. 3. REAR VIEW OF LENTZ ENGINES IN THORP ST. STATION, PEOPLE'S GAS LIGHT & COKE CO., CHICAGO

duction of a horsepower-hour under these conditions with 19.2 lb. of steam means a Rankine efficiency of over 73 per cent. and a thermal efficiency of 12.6 per cent., crediting the engine with the heat which could be taken up by the feed water at the temperature of the engine's exhaust.

The engine, which is 15x21 in., ran at 175 r.p.m. and developed on an average of 164.1 hp. during the three hours of the test. The diagram shown in Fig. 2 was taken about the middle of the test.

Dundee's New Engineering Laboratories

The new engineering laboratories at University College, Dundee, are now in use, though they have not been formally opened, says *Canadian Engineer*. The block is two stories in height. On the ground flood there are a hydraulic laboratory, small lecture room, library, photographic rooms, and engines and generating plant. On the second floor are accommodated the principal lecture room, the drawing offices, the staff rooms, the workshop, and the stress and cement-testing laboratories. Water is supplied to the hydraulic laboratory from a tank at the top of a tower, about 60 ft. above floor level, through an 8-in. main, and is returned to the tank by a motor-driven centrifugal pump which lifts 475 gal. per min.

The heat engine laboratory is divided into an engine room proper and a boiler and producer house, and the engine plant consists of a horizontal tandem-compound engine, with drop and Corliss valves and surface condensers, capable of developing 150 b.hp. at 100 r.p.m. There are also a Babcock & Wilcox boiler with an auxiliary electrically driven feed pump and economizer, the boiler being fitted with a separately fired superheater, and working up to 250 lb. per sq.in.; a 30-hp. gas engine with suction gas producer; a motor-car engine with appliances for quantitative measurements of its power and heat efficiencies, and a 50-ton testing machine.

The Micro Indicator

With increasing speed it became necessary to restrict the movements of the pencil lever and the paper drum of the indicator so as to make smaller diagrams. Some makers of indicators furnish small drums for high-speed work. This means of avoiding distortion due to inertia has been carried to a limit in the micro-indicator invented by a German by the name of O. Mader. This instrument traces on smoked glass, a minute diagram which is magnified 40 diameters by means of a microscope for measurement and analysis, or may be reproduced upon an enlarged scale photographically, or by means of a special reflecting and magnifying device. It is said that the instrument may be used upon internal combustion engines running at a speed as high as 3000 revolutions per minute.

Engineering Progress in U. S. Navy*

SYNOPSIS—Improvements in turbine and engine installations, the more important resulting from higher steam pressure, change in design of frame, reduction of clearance in cylinders, shorter steam ports, increased ratio of expansion of steam and the adoption of forced lubrication.

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Turbine Changes to Produce Increased Economy

The Parsons turbine as it exists in the Navy vessels today is, with very few exceptions, the same as the turbines of this type which were fitted in the initial turbine vessel, the "Chester." The only improvements which have been made consist of changes in blade angles, particularly in the low-pressure stages, an increase in the number of rows of blades in these same stages, and the fitting of nozzles for the admission of auxiliary exhaust steam at several different locations along the steam path.

With the impulse turbine the advance over the original naval turbines of this type in the "Salem" has been rapid.

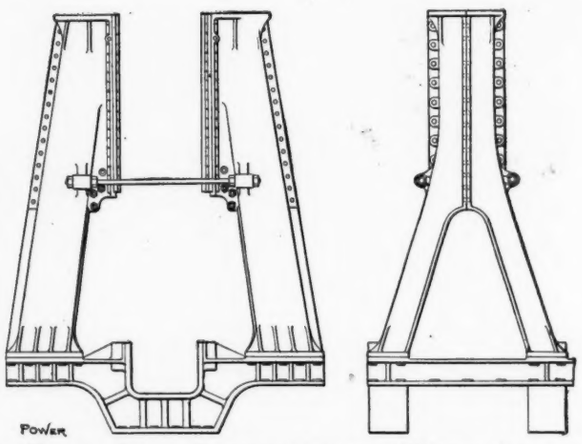

Fig. 1. Older Type of Framing Which Gave Trouble

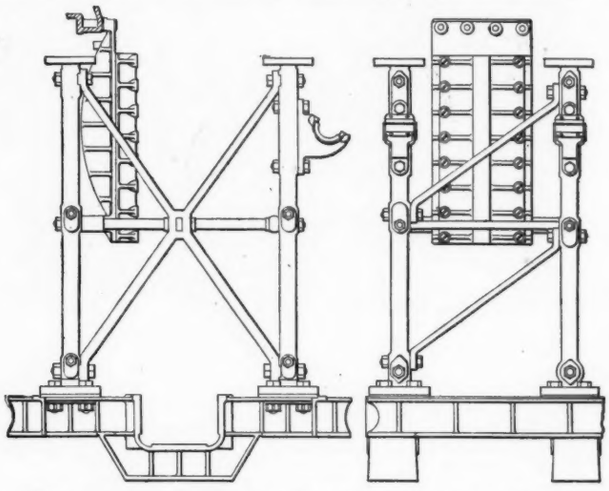

Fig. 2. New Type of Framing Adopted for the "Kearsarge" and "Kentucky"

The number of stages has been very much increased, both in battleship and in destroyer turbines, a drum construction has been adopted for the lower-pressure stages, steam balance for propeller thrust has been provided, cruising nozzles for low fractional powers have been fitted, and nozzles for utilization of auxiliary exhaust are now supplied as in the Parsons turbines.

That these changes in turbines of the impulse type have been accompanied by increase in economy has been thoroughly demonstrated by experience with the machinery of the destroyers, the economy of the impulse turbine showing up nearly, if not fully, as good as that of the reaction type. No opportunity has as yet been offered to obtain a measure of this economy increase with the battleship types of impulse turbine, nor will such opportunity occur until the "Nevada" is ready for trial.

Improvements in Reciprocating Engines

The steps taken in pursuit of the above object are:

1. Increase in steam pressure at engine.
2. Change in design of engine framing.
3. Increase in piston speed.
4. Use of superheat, but to a small degree only.
5. Reduction of clearances in cylinders.
6. Decrease of frictional losses through steam ports.
7. Positive circulation of steam through steam jackets.
8. Reduced back pressure in low-pressure cylinders.
9. Increased ratio between low-pressure and high-pressure cylinders, with consequent increased ratio of expansion of steam.
10. Application of forced lubrication to all journals, crosshead guides, eccentrics and thrust bearings.

The following improvements, both with reciprocating engines and with turbines, have been made:

11. Improved condensing apparatus resulting in higher vacuum.
12. Rational designs of feed heaters based upon the amount of water to be heated and the amount of auxiliary exhaust steam available for heating purposes instead of using the old rule of thumb of allowing a fixed horsepower per square foot of heating surface.
13. Basing steam-pipe design upon actual rate of flow of steam through the pipes as determined by tests in service.
14. Reduction of feed-pipe losses to a minimum.
15. Improved evaporators and other auxiliaries.

In addition to the above, the reliability of the machinery plant has been improved by—

16. Adoption of high-speed, electric-driven forced-draft fans for battleship.
17. Turbine-driven forced-draft fans for destroyers, and the most important of all—
18. The adoption of oil fuel for both battleships and destroyers.

Considering the above changes in detail, the steam pressures at the main engines since 1895 have been increased gradually from 150-lb. gage to 265-lb. gage in the high-pressure valve chest, resulting in decreased size of engine cylinders and in decreased size of steam piping for equal units of power.

The engine framing of the vertical engines first fitted

*From paper read by Captain C. W. Dyson before the Society of Naval Architects and Marine Engineers. First part of the paper abstracted in Power of Jan. 7.

was either of cast steel or built up of steel plates, as shown in Fig. 1. On several vessels trouble has been experienced with this type of framing, particularly when made of cast steel, and, in addition, the weight was high. The Bureau of Steam Engineering, in order to overcome these faults, designed and adopted the style of framing shown in Fig. 2 for the "Kearsarge" and "Kentucky," and this style of forged-steel, built-up framing has been adhered to.

Since the adoption of this type, framing troubles are unknown notwithstanding the fact that the weight of the modern framing per indicated horsepower has been reduced to about 3.3 lb. against 5½ lb. for the old.

Since the design of engines for the "Oregon" was laid down, there has been a gradual increase in piston speeds used, from 900 ft. in that class to 1000 ft. in the "Delaware" class. This increase in piston speed has been followed by a decrease in weight of the moving parts and has aided in holding down the weight and height of the engine, although the stroke has been increased from 42 to 48 in.

In Fig. 3 are shown lines giving the maximum and minimum heights of naval engines for different lengths of stroke, also the line of crosshead pin centers and crankpin centers. These curves are all for engines having the bureau's standard ratio of four to one between the connecting-rod length and crankpin arm

In the use of superheated steam, the Bureau of Steam Engineering has been rather conservative; at present there are seven vessels in the naval service fitted for superheat, the maximum degree of superheat obtained at the boilers being 85 deg. F. which reduces to about 60 deg. F. at the engines. These figures are for full-power conditions, and an increase in economy of about 6 per cent. is estimated to be obtained. At 12 knots, the cruising speed, the saving by the use of superheat hardly exceeds 3 per cent.

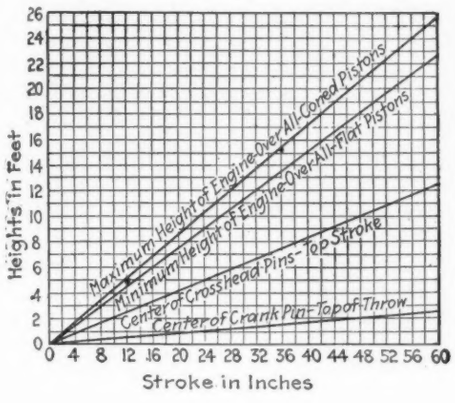
Heights in Feet
Maximum Height of Engine Over All Cored Pistons
Minimum Height of Engine Over All Flat Pistons
Center of Crosshead Pins - Top Stroke
Center of Crank Pin - Top of Throw
Stroke in Inches

FIG. 3. APPROXIMATE HEIGHT OF MARINE ENGINES

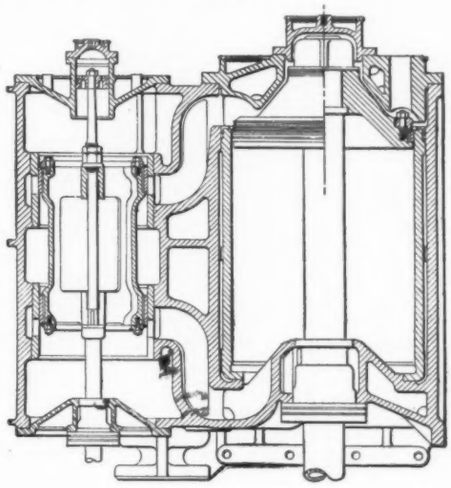

FIG. 4. H.P. CYLINDER U. S.S. "MARYLAND." CLEARANCE 27 PER CENT.

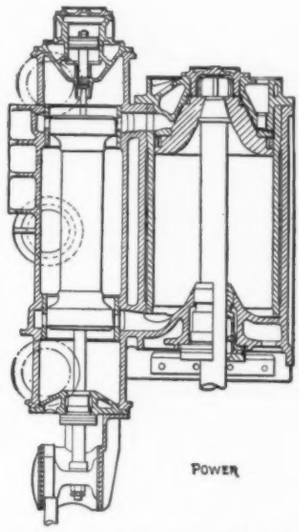

FIG. 5. H.P. CYL. U. S.S. "OKLAHOMA." CLEARANCE 10.75 PER CENT.

The first experiences with the vessels fitted with superheat were far from satisfactory, due to the rapid deterioration of the valves in the steam lines. These valves had cast-steel bodies and cast-steel valve disks with monel metal seats. The erosion and corrosion of the valve disks was extensive, and in a short period of service it became necessary to replace the cast-steel valve disks with disks of monel metal. This substitution has been satisfactory and no further trouble has been experienced. The superheat has been used only on battleships fitted with reciprocating engines or impulse turbines for propelling purposes and has not as yet been used on any of the destroyers.

Reduction of clearances, decrease of frictional resistances of steam through the steam ports and reduced back pressures in the low-pressure cylinders have all resulted from one very important change in the design of

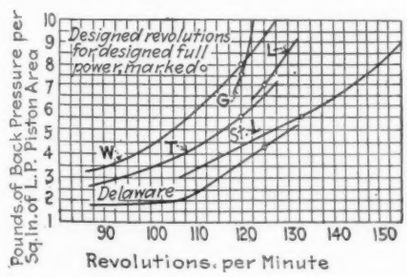
Pounds of Back Pressure per Sq. In. of L.P. Piston Area
Designed revolutions for designed full power, marked o
Delaware
Revolutions per Minute

FIG. 6. BACK-PRESSURE CURVES

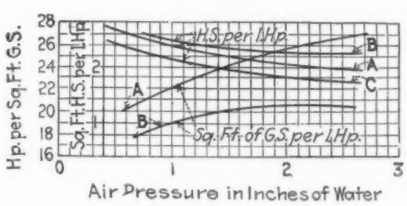

FIG. 7. HEATING AND GRATE SURFACE PER I.HP.

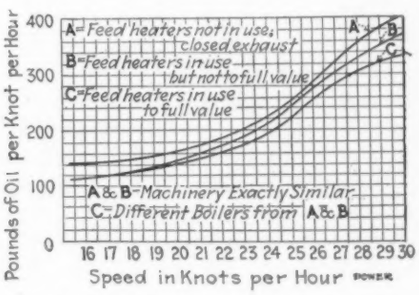
Pounds of Oil per Knot per Hour
A - Feed heaters not in use, closed exhaust
B - Feed heaters in use but not to full value
C - Feed heaters in use to full value
A & B - Machinery Exactly Similar
C - Different Boilers from A & B
Speed in Knots per Hour

FIG. 8. INCREASE OF OIL FUEL WITH SPEED

engine cylinders and valve chests. The original design of these cylinders is shown in Fig. 4, and the modern improved design in Fig. 5.

The results obtained by this change from the long, tortuous port to the short, direct port can best be shown by the accompanying table of comparison of cylinder clearances and steam velocities:

CYLINDER CLEARANCES AND STEAM VELOCITIES

	Old type	New No. 1	New No. 2	New No. 3
Diam. of cylinders, in.:				
H. P.	32½	38½	32	35
I. P.	53	57	52	59
L. P.	2–61	2–76	2–72	2–78
Stroke of pistons, in.	48	48	48	48
Per cent. clearances:				
H. P.	28	16.17	13.83	13
I. P.	20.8	13.17	12.65	13.5
L. P.	18.5	12.485	12.02	12
Steam velocity, ft. per min.:				
H. P.	5,670	6,565	6,723	6,402
I. P.	8,262	6,678	7,883	7,517
L. P.	10,833	10,446	9,947	10,281
Exhaust velocity:				
H. P.	5,480	5,340	5,289	5,078
I. P.	6,670	6,180	5,909	6,087
L. P.	7,450	7,514	7,199	7,298
To condenser	7,390	6,637	6,612	6,500

To further illustrate the gains made by the short direct ports, attention is called to Fig. 6, where the back pressures on the low-pressure pistons of several vessels at varying revolutions are shown in the curves, the designed revolutions being marked on each curve by a circle. Of these vessels, the "Delaware" is the only one with the modern type of port.

In the last ten years, the cylinder ratio of low-pressure to high-pressure, for triple expansion engines, has been increased from about 7:1 to 10:1, including clearances. This increase in ratio had been used previously in remodeling the engines of the "Cincinnati" and "Raleigh," with most excellent results. A serious mistake was made, however, in counting too much on the increased expansions obtained by fitting a smaller high-pressure cylinder than that originally installed, the steam pressure having been increased. The new high-pressure cylinder was made 24 in. in diameter and the ratio of low-pressure to high-pressure cylinder changed to about 11½ to 1. While the economy obtained with these engines was most excellent, the high-pressure cylinders were entirely too small and the engines have never developed the expected power.

By the adoption of forced lubrication for the main propelling engines, the engine friction has been enormously reduced. All the journals are oil born so that no metal-to-metal contact occurs. The result has been that the amount of adjustment and overhaul of the main engines has been decreased to a large extent, and the men who would have been used for this overhaul work can now be used on the auxiliary machinery, to good advantage. This decrease in wear of the bearings, and the cushion provided by the oil, has resulted in a much better maintenance of alignment of the engines, has reduced shocks on the machinery and has reduced vibration due to these shocks. In addition, there is considerable saving in oil at ordinary speeds. At high speeds there still exists a heavy loss of oil, due to splashing on the cylinder heads and also to loss by evaporation from the hot surface of the lower heads.

When first fitted, the forced lubrication gave trouble, due to the oil being drawn through the low-pressure piston-rod stuffing boxes. To remedy this defect, stuffing boxes fitted with steam seals have been supplied, and later reports indicate that where the steam seal is properly fitted no trouble of this kind now exists.

That the foregoing changes have produced great economy is amply demonstrated by the results obtained with the machinery installations of the "Michigan," "South Carolina" and "Delaware." Fig. 7 shows curves of indicated horsepower per square foot of grate surface, and curves of square feet of heating surface per indicated horsepower for vessels fitted with the new type of engines and with boilers having superheaters, and also for vessels with the older type of engines and no superheat.

In laying down these curves, the actual generating heating surfaces and superheating surfaces have been retained, and the grates have been increased or decreased in order to have a constant ratio of heating to grate surface of 48.5, this being the ratio for the boilers of the "Minnesota." At the same time, the air pressures have been corrected inversely as the ratio of the new grates to the actual grates. The results obtained are shown as follows:

Curves of indicated horsepower per square foot of grate surface are:

A—With moderate superheat and modern engines.

B—No superheat and unimproved engines.

Curves of square feet of heating surface per indicated horsepower are:

A—Generating and superheating surface per indicated horsepower with modern engines.

C—Generating surface per indicated horsepower with modern engines and superheat.

B—Generating surface per indicated horsepower with unimproved engines and no superheat.

To obtain high vacuums, the Parsons vacuum augmentor is used extensively in the naval service. In some vessels in the service, in place of the ordinary air pump with augmentor, air pumps of the dual type, as manufactured by Weir, have been fitted, while in other vessels both wet- and dry-air pumps have been used.

Of these systems, that with augmentors and also the dual type appear to give the greatest satisfaction in service, and in addition require less weight and space than the wet and dry system. Abroad, a new system, known as the "Kinetic," has been developed, and all reports received concerning it have been very favorable, but no example of this system yet exists in the American naval service. In conjunction with these improved pump systems has occurred an improvement in the tube spacing and the baffling of the condensers in order that a better separation of air from the water of condensation will occur in the condensers.

The improvements in design of feed heaters, steam pipes and feed pipes naturally followed on the measurements of water consumptions of the machinery taken during the acceptance trials of the vessels. These measurements placed in the hands of the Bureau of Steam Engineering data of great value, and that bureau has attempted to use the full value of it in proportioning these important items. For instance, the feed heaters of the "Delaware" were, for lack of data, proportioned on the basis of so many indicated horsepower per square foot of heating surface, and the two heaters combined have a total heating surface of 2100 sq.ft. In her sister ship, the "Utah," the same degree of feed heating is obtained with heaters having a total surface of only 512 sq.ft.

In the search for economy, the Bureau of Steam En-

gineering has adhered strenuously to the use of feed heaters with auxiliary exhaust steam as the heating medium, using any excess of exhaust in the low-pressure turbines or the second receivers of triple-expansion reciprocating engines. This utilization of the auxiliary exhaust has not been to the taste of the turbine manufacturers who prefer to use all of this steam in the turbines, depending for feed heat upon that derived from steam drains discharging into the feed tanks.

That the bureau's contention that greater economy of fuel is obtained by the utilization of the auxiliary exhaust in the feed heaters is correct is shown by the curves in Fig. 8. The boilers of the vessels from which these results were obtained were so similar that the efficiencies may be taken equal, and the comparison can then be made on an assumption of 100 per cent. efficiency without much error. The fuel oil per knot is obtained from the heat units per knot absorbed by the boilers for the different speeds, and dividing these different heat unit values by an assumed British thermal unit value per pound of fuel. At 29 knots it will be seen that the vessel making full use of her feed heaters required only 84.2 per cent. as much fuel as the vessel putting all the auxiliary exhaust into the turbines (closed exhaust), and only 92 per cent. as much as the vessel using the feed heaters to a limited extent.

In evaporators the improvements consist mainly in the adoption of double effect connecting and in throwing open the gates to other than the standard bureau design although these are not the only changes from former practice. Evaporator feed heaters using the vapor from the evaporators as a heating medium have been fitted, and vapor pipes, better designed for the amount they have to carry, are installed.

Until the adoption of electric driven blowers for battleships and other large vessels and of turbine-driven blowers for destroyers and small vessels, the successful outcome of any heavy forced-draft run was always endangered by the unreliability of the blowers. Since the adoption of these types of blowers this danger of breakdown has been almost entirely eliminated, and so far as the destroyers are concerned, the blowers may be classed as one of the most reliable of the auxiliaries used in naval vessels.

What Captain Dyson had to say concerning oil fuel will be published in a later issue.

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What Is a Horsepower?

By William Kent, M. E.

SYNOPSIS—The author remonstrates against the proposal of the U. S. Bureau of Standards to make the horsepower the exact equivalent of 746 watts, and shows that it involves either the use of an equally "inconvenient relation" in terms of foot-pounds or a change in the well established value of the pound.

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Circular No. 34 of the Bureau of Standards defines a horsepower as 746 watts, and says its value in foot-pounds per second is not a constant but varies with latitude and elevation. The following is offered as a general discussion of the subject. We quote from the circular:

"The usual gravitational value for the English horsepower 550 ft.-lb. per sec., when reduced to watts gives a different number according to the value of the force of gravity employed in the conversion, and hence we find different values in various reference books."

The values cited range from 746, given in many books, down to 745.6 (Standard Handbook for Electrical Engineers, 1910, p. 21), Hering's Conversion Tables (1904, p. 81) give 745.650, and the Bureau of Standards Tables of Equivalents (1906) 745.6494.

"When the horsepower is taken as a specified number of foot-pounds per second, the amount of power represented by it varies for different places, since the weight of a 'pound,' as a unit of force, varies in value as g, the acceleration of gravity, varies. A unit of power, the magnitude of which varies from place to place cannot be acceptable, permanently as a unit." "Some writers prefer to define a pound weight as the weight of a pound mass at a particular latitude, such as 45 deg. (or some prefer the latitude of Paris). While this would make the pound of force a definite quantity, it is nevertheless open to the objection that a pound mass would in general have either more or less than a pound weight, depending on its location."

It may here be said that this objection applies to any system of weights and measures whatever, no matter what values may be given to the units. If a pound mass is a definite quantity of matter, and a pound weight is defined as the force which gravity exerts on that pound of mass then the weight will vary, as gravity varies, with the location. But if we define the unit of force with precision as the force with which gravity attracts a pound of matter at 45 deg. and the sea level, where $g = 32.1740$, and define a foot-pound as the work done by that force acting through a distance of one foot, and the horsepower as 550 of such foot-pounds per second, we have a precise value of the horsepower in English units, which has a definite relation to the watt, and which can be computed when we have the precise values of the meter and kilogram in English units.

"An uncertain definition of the horsepower might cause misunderstandings. No such confusion is possible if the horsepower is defined in such a way as to represent the same amount of power at all places."

No such confusion is possible if we define the horsepower as 550 ft.-lb. per sec., and if we define the foot-pound with precision as above described, the pound in the foot-pound being the force exerted by gravity on one pound of matter at 45 deg. and the sea level, or the force which, constantly applied to a pound of matter free to move, will give it an acceleration of 32.1740 ft. per sec. per second.

The Bureau of Standards neglects this easy method of arriving at a precise definition of the horsepower, which may be valued either in foot-pounds per second or in watts, abandons its former definition, the time-honored 550 ft.-lb. per sec., and its equivalent as formerly computed by the bureau, 745.6494 watts. It says:

"On account of well established use of 45 deg. as a standard latitude, the horsepower tables in this bureau's

Tables of Equivalents formerly assumed 550 ft.-lb. per sec. as the correct equivalent at 45 deg. latitude, which gave the inconvenient relation, 1 hp. = 745.6494 watts. Further editions of bureau publications will use the relation 1 hp. = 746 watts."

The inconvenient relation, it will be noticed, is carried out to the fourth decimal place, or to seven significant figures, indicating an order of precision far greater than that of the value of 39.37 in. in a meter, or than that of any determination of the acceleration due to gravity at any location, both of which values are used in converting foot-pounds per second into watts. The "inconvenience" might have been eliminated even with all practicable scientific precision, by changing the value less than one part in a million, adding 0.0006 watt, and making it 745.650, as in Hering's conversion tables.

Instead of doing this the bureau arbitrarily changes the value to 746 watts, stating that this is equivalent to 550 ft.-lb. per sec. at latitude 50 deg. (instead of 45 deg.) and makes the number of foot-pounds per second in a horsepower at latitude 45 deg., 550.28 instead of 550. It also gives a series of short tables, showing the value of the English and American horsepower (746 watts) in foot-pounds per second and in kilogram-meters per second, at various latitudes and altitudes. From these tables the following values are taken:

Lat. 0 deg. (equator) sea level 551.75; 10,000 ft. alt. 552.08
Lat. 45 deg. (equator) sea level 550.28; 10,000 ft. alt. 550.61
Lat. 90 deg. (pole) sea level 548.82; 10,000 ft. alt. 549.15

Approximate values: London 50 deg., 550; Washington 39 deg., 550.5; New Orleans 30 deg., 551. Taking the figure 550.28 as the value of the horsepower of 746 watts at lat., 45 deg. gives 550.28 : 550 : : 746 : 745.620.

The last figure represents the value of the horsepower (550 ft.-lb. per sec. at lat. 45 deg.) in watts, if we take the bureau's figure 550.28 as the correct figure corresponding to 746 watts. It does not correspond with the bureau's old figure 745.6494. It would appear that a new figure for the acceleration due to gravity at lat. 45 deg. was taken in computing the bureau's new tables.

In order to escape the "inconvenient relation" expressed by the figure 745.6494 and to obtain the more convenient figure 746, the bureau would put engineers to the inconvenience of looking up the tables showing the variation of the horsepower (when expressed in foot-pounds per second) every time they wished to convert foot-pounds per second into horsepower. It would have been much simpler if they had defined the horsepower as 745.62 watts, and stated that it was equivalent to 550 ft.-lb. per sec., giving the foot-pound the precise definition of the work of overcoming a force of one pound through a distance of one foot, and defining the unit of force, as before stated, as the force exerted by gravity on one pound of matter at 45 deg. at the sea level.

The bureau defined the watt as follows: "The watt is the power developed by the action with a velocity of one meter per second of a force capable of giving to the mass of one kilogram in one second a velocity of one meter per second."

Since force is defined (in the metric system) as equivalent, numerically, to the product of the mass (quantity of matter) in kilograms and the acceleration in meters per second per second, and taking g as the acceleration at any given place, the above definition may be converted into the following:

The watt is the power developed by the action with a velocity of one meter per second of $\frac{1}{g}$ of the force capable of giving to the mass of one kilogram an acceleration of g meters per second per second or more simply:

The watt is the power developed by the action with a velocity of one meter per second of $\frac{1}{g}$ of the force which gravity exerts upon the mass of one kilogram, at 45 deg. at the sea level, $g = 9.80665$, the value established by international agreement (Troisieme Conf. des poids et mesures, 1901, pp. 66-68; Marks & Davis' "Steam Tables and Diagrams," p. 93).

Taking this value of the watt and 550 ft.-lb. per sec. as the value of the horsepower, with one kilogram = 2.204622 lb. and one meter = 39.37 in., we obtain the relation between the watt and the horsepower as follows:

$$Power = rate\ of\ work = force \times velocity$$

$$1\ watt = \frac{1}{9.80665} kg. \times 1\ m. \quad per\ sec. = \frac{1}{9.80665}$$

$$\times 2.204622 \times \frac{39.37}{12} = 0.73756046\ ft.\text{-}lb.\ per\ sec.$$

1 ft.-lb. per sec. = 1.35582105 watts
550 ft.-lb. per sec. = 745.70158 watts
1 hp. = 745.7 watts

The error, due to leaving off all the decimals but one, is less than 1 part in 472,000.

For all ordinary engineering work the engineer will use 746 watts as the equivalent of a horsepower of 550 ft.-lb. per sec., just as he uses 32.2 for the value of g instead of 32.1740, the more accurate figure (at lat. 45 deg.), and 778 ft.-lb. as the mechanical equivalent of the British thermal unit lnstead of 777.536, which is computed from Smith's value of the mean calorie 4.1834×10^7 ergs. (*Monthly Weather Review*, October, 1907), $g = 9.80665$, 1 kg. = 2.204622 lb., and 1 m. = 39.37; but when he wishes to have his results computed with accuracy beyond the third significant figure he should use the more accurate constants or tables based upon them.

If we should accept the bureau's advice and adopt 746 watts as the precise equivalent of the horsepower, and consequently give a variable value to the foot-pound, $\frac{1}{550.28}$ of a horsepower at 45 deg. lat., and sea level, and $\frac{1}{550.61}$ of a horsepower at 45 deg. lat., and 10,000 ft. altitude, etc., no end of confusion would result. For example:

A steam engine is tested at New Orleans, lat. 30 deg., and the following data are obtained:

Area of cylinder	330 sq.in.
Steam pressure	150 lb. absolute
Back-pressure	18 lb. absolute
Mean effective pressure	50 lb. per sq.in.
Piston speed	600 ft. per min.
Feed water temperature	200 deg. F.
Feed water consumption	12,000 lb. per hr.

Required the indicated horsepower, the steam consumption per horsepower-hour, and the thermal efficiency.

DATA FROM STEAM TABLES

B.t.u. per lb. steam, 150 lb.	1193.4
B.t.u. per lb. water, 200 deg. F.	167.9
Difference	1025.5

Computed results, engineers' method:

$$Hp. = \frac{PAS}{33{,}000} = \frac{50 \times 330 \times 600}{33{,}000} = 300\ i.hp.$$

Steam per i.hp. per hr. 12,000 ÷ 300 = 40 lb.

Heat equivalent of a hp.-hr. $= \frac{1,980,000}{777.5} = 2547$ B.t.u.

Heat expended per hp.-hr. $1025.5 \times 40 = 41,020$ B.t.u. Thermal efficiency $\frac{2547}{41,020} = 6.209$ per cent.

According to the Bureau of Standards methods. we would find from the bureau's tables that at New Orleans a horsepower is approximately 551 ft.-lb. per sec., and that a pound force at New Orleans is $\frac{550}{551}$ of the pound force at lat. 50 deg. Then we would have to consider what is meant by a pound in the data, 150 lb. absolute pressure at 50 lb. m.e.p. The first is measured by a pressure gage and the second by the spring of a steam-engine indicator. The first may have been standardized by dead weights in Washington and the second by dead weights in Boston and the standardization may or may not have been reduced to the equivalent at lat. 45 deg. or 50 deg. The figures taken from the steam tables are based on the fact that a pound pressure is equivalent to the force exerted by gravity upon a pound of matter at 45 deg. at the sea level or at any place where the acceleration due to gravity is 32.174 ft. per sec. per sec. The constants used in the engineers' computations, 1193.4, 167.9, 1,980,000, 777.5 all are referred to a pound of force having a definite and constant value. Do any of these figures, or do any of those relating to pressures in the observed data need to be multiplied or divided by the ratio $\frac{551}{550}$ or by the ratio $\frac{551}{550.28}$ in order to obtain correct values for the horsepower, the steam consumption per horsepower per hour, or the thermal efficiency? The question may be referred to the bureau.

According to the bureau, the watt is the product of a definite force into a definite velocity. The force is $\frac{1}{9.80665}$ of a kilogram force, or of the force exerted by gravity on one kilogram at lat. 45 deg. The horsepower is defined as 746 watts. Why not define it also as the product of a force into a velocity, or as a convenient multiple (say 550 times) that product, and why not give a definite and precise value to the two factors of that product, viz.: the force in pounds and the velocity in feet per second? The precise definition of a pound force, as that force which will give an acceleration of 32.1740 ft. per sec. per sec. to a pound of matter, is in harmony with our modern steam and entropy tables and with all our conceptions of forces that are other than the force of gravity, such as the pressure of steam, the tensile strength of metals, the reaction of a spring, the pressure required to compress air, etc. This definition of the pound force is in exact harmony with the following definition given in Louis A. Martin's, Jr., "Text Book of Mechanics," vol. IV, p. 1: "The unit of force is the attraction exerted by the earth upon a certain piece of platinum when at a latitude of 45 deg. and at the sea level. This piece of platinum is marked P. S. 1844, 1 lb. and is carefully preserved in London."

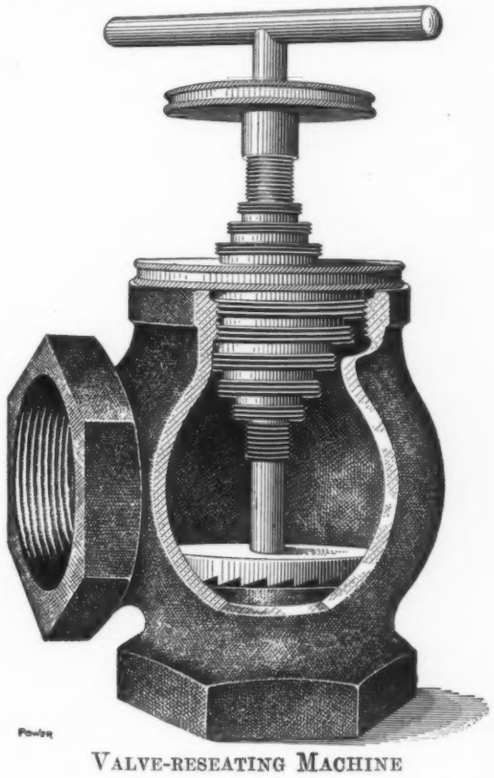
The only excuse given by the bureau for its new definition of the horsepower is that the figure formerly used by it in converting watts to horsepower 745.6494 is an "inconvenient relation," and that 746 is more convenient. The bureau's old figure has a great pretension to accuracy, but it is not nearly as correct (if we accept the international standard value of $g = 9.80665$) as the figure 745.7, which is almost as convenient as 746, and vastly more convenient than the set of variable values 550.28, etc., ft.-lb. given by the bureau.

The bureau's circular contains the following: "It is hoped that engineering societies and other interests concerned will recognize the value of the English and American horsepower as 746 watts, or 550 ft.-lb. per sec. at 50 deg. and sea level (approximately latitude of London), employing Table II to obtain the value in foot-pounds per second at other places."

It is hoped, from the engineer's point of view that the societies and other interests will commit no such blunder, but that on the contrary the bureau will withdraw its circular No. 34, and get itself in harmony with the best modern engineering practice by defining a pound force as the force which will give a pound matter an acceleration of 9.80665 m., or 32.1740 ft. per sec. per sec., and the horsepower as 550 ft.-lb. per sec. 33,000 ft.-lb. per min., which is equivalent to 745.7 watts.

Abbott Valve Reseating Machine

This simple device, manufactured by the Abbott Hardware Co., 636 Columbus Ave., New York City, is illustrated herewith. It consists of a knurled center disk with tapered threaded disks on either side, which fit into the bonnet opening of globe and angle valves, as shown. This tool is made for valves ranging from ⅛ up to 2 in. in size. Cutters are supplied with each machine suitable for different sized valves.



VALVE-RESEATING MACHINE

To regrind a valve seat, the tool is screwed into the valve after fitting the proper size cutter to the spindle. The cutter is then turned by the handle, and the feeding is done by the top knurled handwheel, which prevents chattering of the knife, and insures a smooth seat.

The Augustine Rotary Engine

In a rotary engine of the type shown in Fig. 1 it is evident that the pressure in the active part, 31, of the cylinder will not only force the piston, 8, ahead, but will exert an equal force per unit of area upon that portion of the surface of the cylinder, 4, which lies between the running joint, 13, and the piston. As this cylinder is carried upon the main shaft, an excessive pressure, varying in magnitude and direction, is exerted upon the bearings. One of the modifications which the Augustine rotary engine has undergone since our description of it in the Dec. 21, 1909, issue is designed to overcome this difficulty.

If, as in Fig. 2, a second cylinder, 18, concentric with the piston cylinder, 4, be carried upon the same shaft and small apertures, 30, be provided to transmit the pressure existing in the chamber, 31, to the pockets, 14, formed by ribs, 16 (see also Fig. 5), upon the outer surface of the intervening abutment, 3, the pressure in the acting cylinder will be transmitted to the inner surface of the balancing cylinder, 18, and the shaft be lifted thereby in a contrary direction to that in which it is forced by the pressure upon the piston cylinder, 4, relieving the pressure upon the bearings.

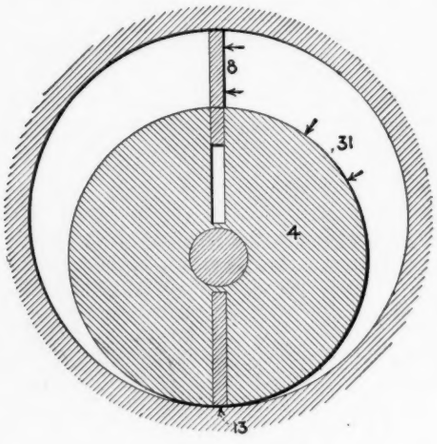


Fig. 1

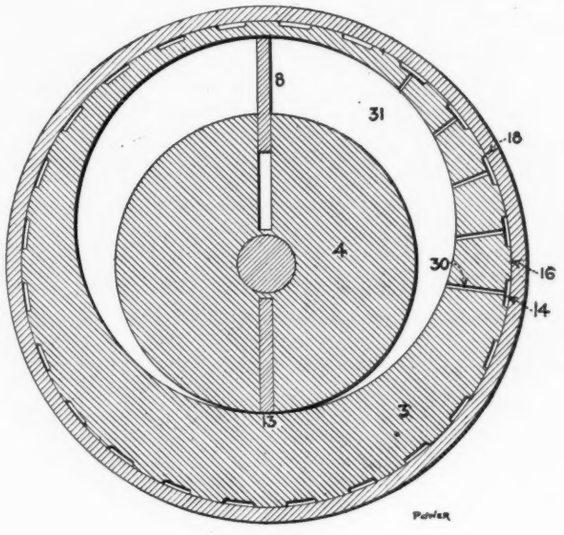


Fig. 2.

Showing Thrust on Bearings in Rotary Engine

How this idea has been applied to the engine is shown by the accompanying engravings. Fig. 3 shows the piston cylinder, 4, with the pistons, 8, and one portion of the balancing cylinder, 18. A similar portion slips over the nearer head and the tongued inner edges of these balancing cylinders run in the grooved edges of the ring, 15, upon the center of the abutment, Fig. 5. The apertures leading from the cylinder to the pockets have check valves to prevent the return of the oil which is forced through them by centrifugal force and which creates a film preventing steam leakage. This central ring, 15, fits tightly into the casing, 2, Fig. 4, to which it is firmly bolted from the outside, and the balancing cylinders, 18, Fig. 3, run in the annular spaces, 19, Fig. 4, at either end. Openings in the ring, one of which is shown at 20 in Fig. 5 permit the steam to enter and escape. The admission ports are shown at 21 and the exhaust ports at 22 in Fig. 4. The joint, 13, in which the cylinder, 4, Fig. 3, runs is turned to the radius of that cylinder and provided with water grooves or pockets. That the balancing is effective may be demonstrated by removing the caps from the journal bearings when the engine is running. The shaft will be found to be floating freely.

The exhaust is valveless, taking place when the piston passes the upper ends of the ports, 22, Fig. 4. The admission and cutoff are controlled by a cylindrical valve, 6, Fig. 8, which is turned through an arc of a circle by means of the attached toothed segment which gears with a segment of twice its radius on the shaft, 7, which projects as shown in Fig. 4 and carries an arm which is attached to the shaft governor as shown in Fig. 9.

As with increasing speed the governor weights are thrown outward, the throw of the governor eccentric is decreased, the movement of the valve diminished and the point of cutoff made earlier. The multiplication of the movement of the valve by the use of the larger segment gives quick opening and closure.

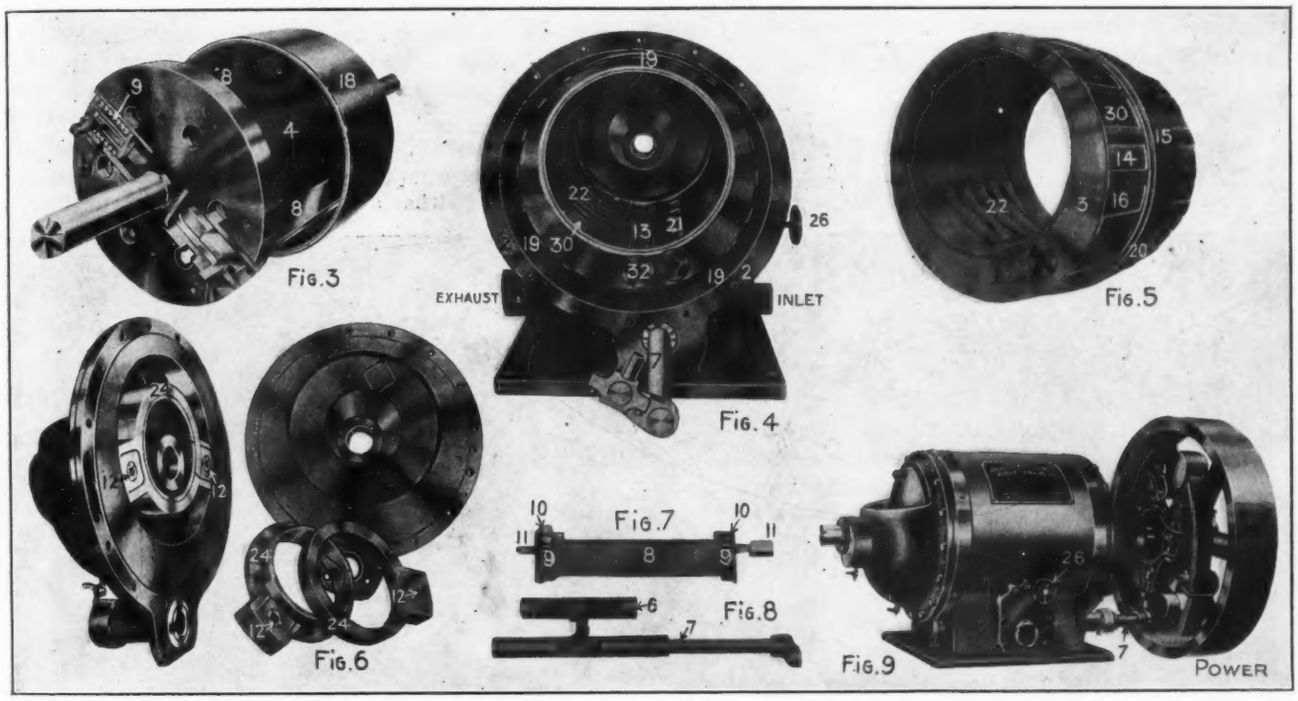
Other important sources of friction and wear in the earlier engine were the movement of the piston blades in and out against a heavy pressure and the scraping of the outer edge of the piston against the inner surface of the cylinder as it was thrown outward against it by centrifugal force. To reduce the first the pistons have been provided with roller bearings at their ends, as shown at 9 in Figs. 3 and 7, and to avoid their pressure upon the cylinder they are carried positively by the blocks, 12, Fig. 6, into which the bushed shanks, 11, of Fig. 7, fit. The centers of the overlapping rings, 24, Fig. 6, being eccentric to the center of the mainshaft, the distance between the holes, 12, and the shaft center changes as they are carried around in their eccentric path, and the pistons are positively carried inward and outward in their grooves with no other contact with the cylinder surface than is necessary for a tight joint.

The plate, 10, Fig. 7, closes the opening in the head of

the piston cylinder in which the pistons slide, as shown in Fig. 3. The spring-backed plug, 32, Fig. 4, runs against the inner surface of the head of the balancing cylinder, 18, Fig. 3, and prevents steam from blowing by the ends of the running joint, 13, Fig. 4, from the high-pressure side of the cylinder to the exhaust. The bearings are carried in the heads, as shown in Figs. 6 and 9, and are ring-oiled. In order to start the engine when it has stopped with the piston beyond the point of cut-off, a bypass valve, 26, Figs. 4 and 9, is provided.

Several hundred of these engines varying in capacity from 5 to 200 hp. and mostly of the earlier design have been built and are at work. The normal speed varies from 400 r.p.m. in the smaller sizes to 300 or 350 for engines of 200 hp., although the inventor claims that the smaller engines may be run as high as 2100 r.p.m. and places no limit upon the capacity for which he will take an order.

The engine shown in Fig. 9 is 19 in. long, 17 in. wide at the base, stands 20 in. high, has a shaft 48 in. long, is 17 in. outside diameter and weighs 800 lb. complete. Tested at the Ohio State University it delivered 39.69 b.hp. on 36.79 lb. of steam per b.hp.-hr., the steam being at a pressure of 106.7-lb. gage superheated 116 deg. and the exhaust into the atmosphere. The engine is built by the Augustine Automatic Rotary Engine Co., Buffalo, N. Y., and the Augustine Rotary Mill & Factory Engine Co., Mt. Vernon, Ohio.



THE AUGUSTINE ROTARY ENGINE. FIGS. 3 TO 9.

Decomposition of Steam by Heat

By DR. AUGUSTUS H. GILL

Among the various means for the prevention of smoke is the jet of steam, used chiefly to suck in a current of air. This it certainly does, but it also gives an opportunity for the advancement of extravagant claims for the device as a gas producer; I remember seeing the legend "Hydrogen Gas Retort" upon a circular of such a device.

One of the claims commonly advanced, is for great economy due to the decomposition of the steam by the hot coal. While this is true according to the reaction

$$\underset{\text{coal}}{\text{Carbon}} + \underset{\text{steam}}{\text{hydrogen oxide}} = \text{carbonic oxide} + \text{hydrogen}$$

yet there can be but little or no economy in it, because heat is used up in decomposing the water. This fact is lost sight of by the advocates of such apparatus. The reaction that has taken place is analogous to rolling a heavy stone up hill. When the enthusiasts over first aids to combustion of this kind point out "the amount of work that can be done if the stone be allowed to roll down;" they appear, as it were, when the stone is at the top of the hill, to forget, and make the victim forget, the toil necessary to get it there.

Similarly they disregard, and make others lose sight of, the energy used up in decomposing the steam. There is practically nothing to be gained by this operation because the heat that the combustion of the hydrogen produced is exactly equal to that used up in the decomposition of the steam. There may be a slight economy in that the carbonic oxide formed may be consumed to better advantage than the coal from which it is made. The temperatures required for the decomposition of steam by coal are from 1245 deg. F., when about 9 per cent. is broken up, to 2057 deg. F.* when it is practically all decomposed.

Another claim, and one even more preposterous, is that by passing steam through white-hot iron retorts or pipes it is decomposed into hydrogen and oxygen, of which one is available as a fuel and the other as an aid to combustion. This probably comes from the fact that when the apparatus is new, hydrogen will actually be found in the gases issuing from it. It is not formed from

*Bunte J. Gas Lighting 53, p. 481.

the decomposition of water into hydrogen and oxygen, but in accordance with the well-known fact that red-hot iron will decompose steam according to the reaction:

iron + hydrogen oxide = iron oxide + hydrogen

This has been made use of for many years for the protection of iron fittings and is called the Bower-Barff process. When, however, the pipes and retorts become thoroughly coated with this oxide of iron, (forge scale, magnetic oxide) no further decomposition of the steam can take place.

The practical difficulties in the way of this claim are:

First, it is impossible to heat steam hot enough in a coal-heated furnace to decompose any usable amount.

Second, if it were decomposed, when it is cooled down, for example to pass the mixture under the grate, the hydrogen and oxygen would recombine, and merely superheated steam result.

Third, at the temperature necessary to decompose any appreciable quantity of steam, the iron (even wrought-iron) fittings would be melted and give way under the pressure.

Fourth, if it were possible to dissociate steam under these conditions there would be nothing gained, for the reason given above, in the decomposition with carbon; the heat absorbed in the decomposition of steam is only just returned in the combustion of the hydrogen and oxygen formed.

Lowenstein and Wartenburg in 1906* showed that at 2060 deg. F., about the melting point of average cast iron (1912 deg.—2310 deg.) steam was but 0.0086 per cent. decomposed; and even at a temperature but little above that of melting wrought iron (2900 deg. F.) only one-third of one per cent. was dissociated.

This means that in cast or wrought-iron retorts practically no steam can be decomposed into hydrogen and oxygen; and if there were any of these gases formed, inasmuch as the reaction by which they are formed is under the conditions a reversible one, that they will recombine the moment the temperature is lowered.

From these considerations it would seem to be clearly impracticable to use the process in boiler practice.

*Zeit. f. physik. Chem. 54, 715.

Visit to the Mesta Machine Shops

The Engineers' Society of Western Pennsylvania visited the shops of the Mesta Machine Co., at West Homestead, on Saturday, Dec. 28.

A shop like that of the Mestas is always full of interest, but upon this occasion there were upon the floor two engines of more than ordinary account; the larger of these was a twin-tandem compound-reversing engine to be installed in the works of the Youngstown Sheet & Tube Co. It has two 46-in. high-pressure cylinders and two 76-in. low-pressure cylinders, with a common stroke of 60 in. The maximum speed is 200 r.p.m., and it is said to be the most powerful rolling-mill engine ever built. The throttle and reverse gears are controlled by a single lever, which has the advantage of simplicity and compels the engineer to run always with the most economical cutoff. The engine also has a more direct drive and a less complicated valve-gear, with fewer parts than are used on the usual engine of the rolling-mill type.

The smaller unit was a direct-flow steam engine to be installed in the power plant of the Carnegie Institute of Technology. It has a single cylinder 28 in. in diameter by 42-in. stroke, and is a modified Stumpf unidirectional-flow engine, having Corliss steam valves in the head and Corliss exhaust valves at some distance toward the center from the head, thus making a pocket in the hot end of the cylinder. It has not the central valveless exhaust port, and the piston is therefore much shorter, resulting in a much shorter cylinder than in the usual Stumpf engine. On account of the lesser compression also a smaller cylinder diameter can be used, and they were able to obtain this modified form of the Stumpf adapted to noncondensing operations, at a cost something like 30 per cent. below that of the full-fledged unidirectional-flow machine. This engine is guaranteed to run on 27 lb. of steam per kw.-hr., the pressure being 175-lb. gage with 100 deg. of superheat and the exhaust into the atmosphere.

The guarantees for noncondensing steam turbines ran from 35 to 47 lb. per kw.-hr. under the same conditions.

We are promised details and illustrated descriptions of these engines as soon as they are in position and have demonstrated what they can do.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA AT MESTA MACHINE CO., WEST HOMESTEAD, PENN.

ELECTRICAL DEPARTMENT

Primer of Electricity

By Cecil P. Poole

Dynamo and Motor Characteristics

Every direct-current machine of regular construction may be operated either as a dynamo or as a motor. There is no difference between the two machines except in their controlling apparatus; used as a dynamo, the machine is provided with a high-resistance rheostat connected in series with the shunt field winding; used as a motor, it is provided with a low-resistance starting rheostat connected in series with the armature.

The speed of a given machine, when operating as a dynamo, must be appreciably higher than that at which it runs as a motor, if the voltages are the same in both cases. The drop of voltage in the armature circuit causes this difference in speed. For example, if a dynamo delivered 110 volts and 100 amp. running at 1150 r.p.m. and the resistance of the armature circuit were 0.05 ohm, the drop in the armature circuit would be

$$100 \times 0.05 = 5 \text{ volts}$$

and the total electromotive force generated in the armature winding would be 115 volts; 110 delivered at the brushes and 5 absorbed by the internal resistance.

If that same machine were operated as a motor on a 110-volt supply circuit and given a load that caused the armature to take 100 amp., the drop in the armature circuit would be 5 volts, as before, but this would be subtracted from the impressed electromotive force of 110 volts, leaving 105 volts to be balanced by counter electromotive force. As the armature generated 115 volts at 1150 r.p.m. when driven as a dynamo, it will generate 105 volts at 1050 r.p.m. This, therefore, will be the speed when running as a motor, with the same armature current as before.

The foregoing statements apply to either a shunt-wound or a compound-wound machine, but with the latter type (to get the same comparison) the connections of the series field winding must be reversed in order that the magnetizing effect of the winding may be the same in both cases. Figs. 121 and 122 will assist in making it clear why this is true. In the dynamo, Fig. 121, the current flows *from the positive brush to the line* through the series winding, exciting that winding in the same direction that the shunt winding is excited. Running as a motor (Fig. 122), the current flows *from the line to the positive brush* through the series winding, exciting the latter in the opposite direction. Reversing the terminals of the series winding, as in Fig. 123, corrects this.

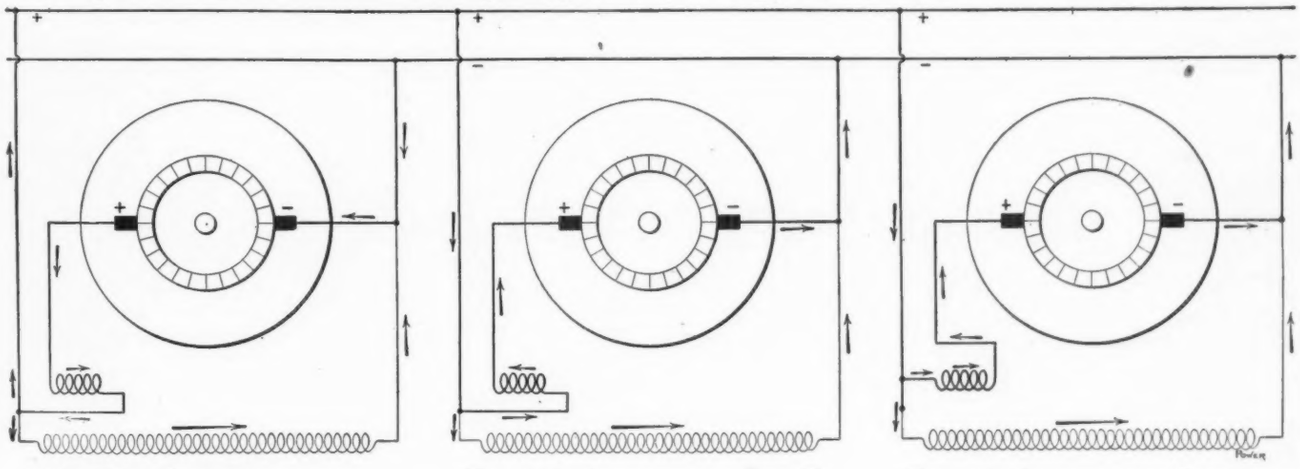


Fig. 121. Compound-wound Machine as a Dynamo

Fig. 122. Compound-wound Machine as a Motor, Connections Unchanged

Fig. 123. Connections Corrected for Running as a Motor

A compound-wound dynamo can be used as a motor without changing the connections of the series field winding, but it is not usually advisable because when this is done the machine becomes a differential compound-wound motor and therefore has a relatively small starting torque, owing to the weakening effect of the series winding on the field magnet. Such a machine also tends to increase its speed as the load increases, and with a very heavy load may "break down" and stop running, due to the reduced armature torque. This could readily happen if the ampere-turns in the series field winding at full load were a large proportion of the ampere-turns in the shunt winding.

For example, if the shunt field winding contained 4000 ampere-turns and the series field winding 2000 (at full load), the total full-load excitation, with the windings magnetizing in the same direction, would be 6000 ampere-turns. Suppose, for convenience, this produced a flux of 6,000,000 maxwells and that the armature torque developed by this flux and full-load current was 300 lb.-ft. Also suppose that 2000 ampere-turns of excitation would produce 3,000,000 maxwells, or one-half of the normal

flux. Then, with the series winding opposing the shunt winding, the net field excitation would be

$$4000 - 2000 = 2000 \textit{ maxwells}$$

and the result would be one-half the correct flux and, consequently, one-half the full-load torque with full-load armature current flowing. If the armature could speed up to twice its normal rate without increasing its load, that is what it would do, because of the halved field flux; and it would deliver the normal full-load work. But if the load were such that it increased with the speed, the motor could not speed up because its torque would not be

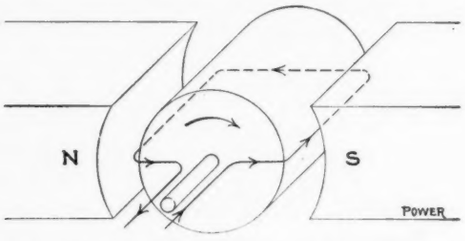


FIG. 124. DIRECTION OF ARMATURE PULL

strong enough to pull the increasing load; it would probably "break down" and stop.

To use a series-wound dynamo as a motor, or *vice versa*, either the terminals of the field winding or the connections at the brushes must be reversed, if the armature is to run in the same direction as before. If the direction of armature rotation be reversed, no change need be made in the connections of the machine.

DIRECTION OF ROTATION

The direction in which a motor armature will revolve when supplied with current depends on the connections between the field winding and the brushes. The reason for this is that the direction in which the current flows through the armature wires *with respect to the polarity of the field magnet* determines the direction of armature torque or pull, as explained in the recent lesson on the electric motor. Thus, if current be sent through the wire loop on the armature core in Fig. 124 in the direction indicated by the arrowheads, the core will tend to turn in the direction of the curved arrow. If the current be reversed in the wire loop, the core will tend to turn in the opposite direction. If the current flow be as indicated by the arrow-heads and the field magnet polarity be reversed, the direction of the armature pull will be reversed.

The direction of the current through the armature wires depends on the direction of entrance and exit at the brushes, and the polarity of the field magnet depends on the direction of current flow through the field winding; hence the truth of the statement at the beginning of the preceding paragraph.

Fig. 125 is a diagram of a shunt-wound motor in which it is assumed that with current flowing through the armature and field windings in the directions indicated by the arrows, the armature will revolve in the direction indicated by the arrow *R*. If the wires leading to the brushes be transposed, as in Fig. 126, the direction of the current through the armature wires will be reversed and the armature will therefore revolve in the opposite direction, indicated by the arrow *R*. Again, if the armature connections be left as in Fig. 125 and the field leads be reversed, as in Fig. 127, this will reverse the field magnet polarity and the armature will revolve in the opposite direction to that in Fig. 125.

If all of the motor connections be left as in Fig. 125 and the leads from the main terminals to the supply circuit be transposed, making the right-hand motor terminal positive and the left-hand one negative, the direction of rotation will not be changed, because both the current in the armature wires and that in the field winding will be reversed and the direction of the current through the armature wires *with respect to the field polarity* will be unchanged.

The foregoing statements regarding the direction of motor rotation apply also to self-exciting dynamos. If the armature of such a dynamo be driven in the opposite direction to that for which it was intended, it will not generate current unless the connections between the field winding and the brushes be reversed. Figs. 128 to 130 show why this is true. If the armature generates an electromotive force in the direction indicated by the arrows *E, E,* Fig. 128, when driven in the direction of the curved arrow and with the field current flowing as indicated, reversing the direction of rotation will reverse

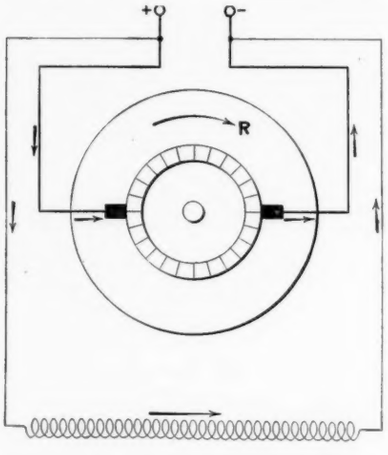


FIG. 125. SHUNT-WOUND MOTOR FOR RUNNING IN NORMAL DIRECTION

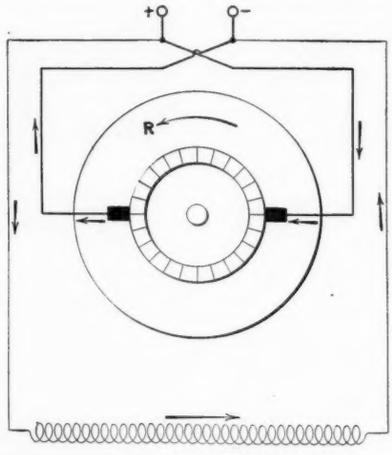


FIG. 126. ARMATURE CONNECTIONS REVERSED FOR REVERSE ROTATION

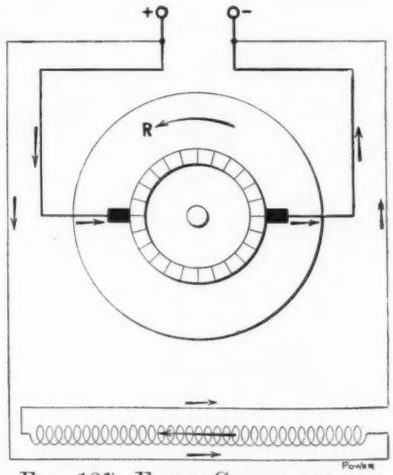


FIG. 127. FIELD CONNECTIONS REVERSED FOR REVERSE ROTATION

the direction of the electromotive force, and, to continue the field excitation in the same direction as before, the connections to the brushes must be reversed, as in Fig. 129.

If the connections between the field winding and the brushes are not reversed, the machine cannot excite itself with the armature rotation reversed, because the direction of the electromotive force generated by the armature will be opposite to the direction of current flow required in the field winding. Fig. 130 illustrates this condition. With current flowing through the field winding in the direction of the arrow and supplied from some other dynamo, the armature electromotive force will be in the direction of the arrows *E, E*. Therefore, if the field winding were connected to the brushes as in Fig. 128, the polarities would be wrong and the armature could not excite the field magnet and consequently could not generate current.

A shunt-wound machine when running as a motor will run in the same direction in which it is driven when operating as a dynamo. Comparison of Figs. 125 and 128 will make this clear. The former shows the direction of current flow when running as a motor. As the counter electromotive force of a motor armature is opposite in direction to the flow of current sent through the armature from the line, arrows indicating the generated electromotive force in Fig. 125 would be as shown at *E, E,* Fig. 128; therefore the latter diagram shows the dynamo conditions for the machine represented as a motor in Fig. 125. Note that the direction of rotation is the same in both cases.

A series-wound machine when operated as a motor will run in the *opposite* direction to that in which it must be driven to operate as a dynamo. This fact is of very little practical importance, however, because it is very seldom that any occasion arises for using a series-wound dynamo as a motor or the reverse. Should such a change be desired, it would be necessary to reverse either the direction of rotation or the connections at the brushes of the machine.

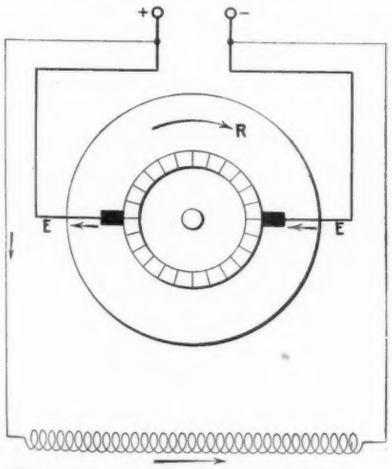


Fig. 128. Shunt-wound Dynamo Normally Connected

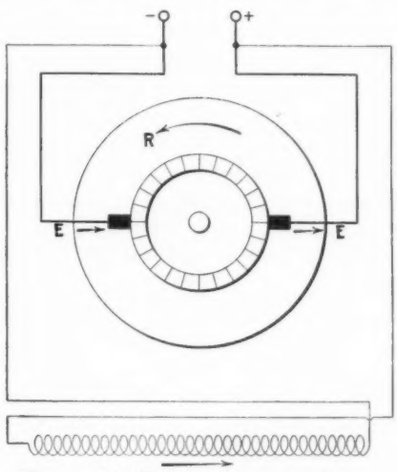


Fig. 129. Field Connections Reversed for Reverse Rotation

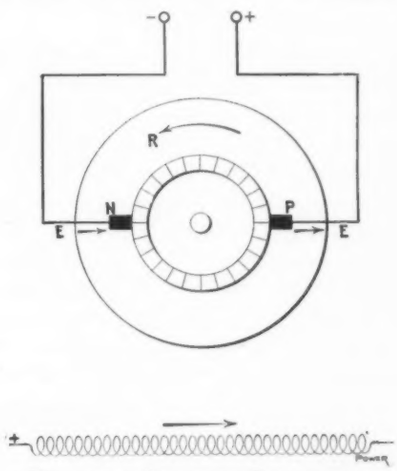


Fig. 130. Reverse Rotation with Separate Excitation

Dynamo "Breakdown" Point

If the load of a shunt-wound dynamo be increased beyond a certain point, the machine will lose its field magnetism and stop generating. This is because the drop in the armature circuit reduces the electromotive force at the brushes to such a point that the field excitation drops below the point of "stability." This carries the reader back for a moment to a consideration of the excitation curve. Fig. 131 is such a curve; the reason for its shape was explained in an early lesson. This curve was plotted from the data of a 220-volt dynamo. The numbers on the horizontal scale at the bottom of the chart are voltages at the terminals of the field winding. The vertical scale shows voltages generated by the armature corresponding to the different excitation voltages. For example, with 90 volts at the field terminals the armature generated 140 volts; with 140 exciting volts, it generated 200 volts, and with 240 volts at the field-winding terminals the armature generated 240 volts.

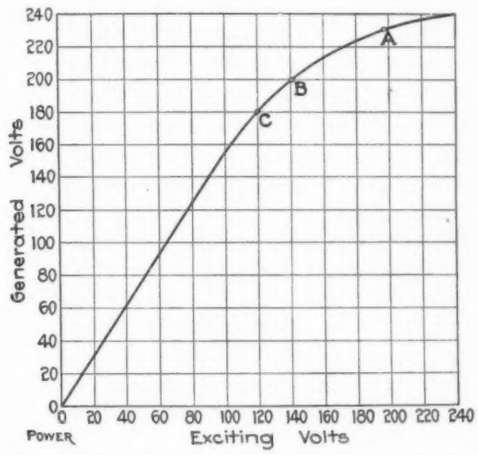


Fig. 131. Excitation Curve of a 220-volt Motor

Now suppose the resistance of the armature circuit to be 0.1 ohm and the drop at the brush faces to be 2 volts at all loads. With 330 amperes going through the armature the total drop would be

$$33 + 2 = 35 \textit{ volts}$$

so that the exciting volts would be 35 less than the gen-

erated volts. This condition exists when the generated volts are 230 and the exciting volts 195; that is, at the point *A* on the curve. Increasing the load to 580 amp. would make the drop 58 volts in the armature plus 2 at the brushes, or 60 total. The exciting volts would therefore be 60 less than the generated volts. The machine might be stable under this condition at the point *B* on the curve, where the generated volts are 200 and the exciting volts are 140, but that is doubtful because it could drop to the point *C* and still have a margin of 60 generated volts above the exciting volts.

Any further increase in load will "break down" the machine—that is, cause it to lose its magnetization and stop generating, because the difference between the exciting and generated volts would be greater than it could stand. For example, if the load were increased until the total drop through the armature and brushes were 65 volts, the machine could not generate enough voltage to excite the field, because there is not a point on the curve where the machine generates 65 volts more than the pressure required to excite the field.

In the example just given, no account was taken of the weakening effect of armature reaction upon the field, because that subject has not yet been considered. The influence of armature reaction merely hastens the "breaking down" of the dynamo, causing it to occur higher up on the excitation curve than it otherwise would.

In practice, a well-designed machine never reaches the break-down point because it is built to work at a point so far up the excitation curve that the armature current would have to be about three times the full-load current to cause enough drop through the armature circuit to make it let go.

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Losses in Turbo-Alternators*

The total iron and copper losses in a large, high-speed turbo-alternator are in general no higher than in a corresponding capacity low-speed machine.

As far as the iron losses are concerned, no further comment need be made than that the magnetic flux densities in general are somewhat lower than in lower speed machines of the same frequency, and therefore the losses per unit volume of material are no larger.

The total armature copper losses in turbo-alternators, as a rule, are considerably smaller than in corresponding capacity machines of the moderate- or- low-speed types. This is due partly to the use of a smaller total number of conductors, and partly to a lower current density in the armature conductors. In a narrow-core machine, a considerable portion of the buried copper heat may be conducted lengthwise of the conductor into the end winding, and there dissipated into the air. In the turbo-generator, with its much wider core and greater distance from the buried copper to the end windings, a smaller percentage of the buried copper heat will be conducted into the end windings.

To partly compensate for this, it is usual to work the armature copper in the turbo-generators at a lower current density, and, therefore, at a relatively lower total copper loss. This is somewhat of a handicap in the economical design of the generator, as extra space is thus required for the armature winding. In some of the earlier machines, the armature conductors were made of solid copper bars of relatively large section, partly for stiffening or bracing the end windings. With these solid conductors there was a very considerable loss in the buried copper, due to eddy currents. To compensate for this, the armature conductors were made very large in section, so that the current density, due to the work current alone, was very low compared with practice in other types of machines. On account of the comparatively large section of armature conductors, the conduction of heat from the buried copper to the end windings was relatively large.

In some of these earlier large capacity machines, the nominal current density in the armature conductors was so low, and the section of conductors so great, that the total buried copper loss, due to the work current, could be carried from the buried part of the coils into the end windings with a comparatively small drop in temperature, so that, if there had been no eddy currents present, the buried copper would have shown less rise than the iron. Any considerable rise which occurred was thus chargeable to eddy currents in the buried conductors, rather than to the work current. While such construction was fairly effective for the purpose, yet it was decidedly uneconomical in design. In fact, with later proportions and methods of design, the safe outputs of some of the earlier machines could easily be 50 to 75 per cent. greater, largely on account of elimination of eddy currents and improvement in methods of dissipating heat from the end windings.

In many of the older machines, the ventilation of the end windings was not nearly as effective as in modern types, due principally to the form and arrangement of the end connectors. Usually air spaces were allowed between adjacent coils, although, in some instances, these were so small as to give but little benefit. Moreover, in many cases, the type of end winding employed rendered these air spaces between coils rather ineffective, unless special means were taken to deflect the air between the coils. With later constructions, the end windings lie more or less across the path of the ventilating air, and there are ample openings between the coils, so that a very considerable part of the ventilating air will actually pass between the coils of the end windings in such a way as to give the maximum possible ventilation. When it is considered that the total armature copper loss may be only 20 per cent. of the total stator loss, it will be seen that an excessive amount of air is not required when the end windings are arranged for most effective ventilation.

Much effort has been expended in eliminating or reducing the eddy current losses in the buried copper of large turbo-generators, as well as in other types of large capacity alternators. These eddy currents are due to two sources, namely, the alternating magnetic flux across the slots, due to the armature ampere turns per slot, and secondly, the magnetic fringing from the rotor pole face into the open armature slots. In some instances, tests have indicated that the local electromotive forces set up in the armature conductors by the flux through the slot opening are much greater than those due to the flux across the slot.

The simplest remedy for the eddy currents set up by these local electromotive forces is to subdivide the conductors into a number of wires or conductors in parallel, so arranged or connected that the local electromotive forces oppose and, to a great extent, balance each other.

*Excerpt from a paper by B. G. Lamme, read before the American Institute of Electrical Engineers, New York, Jan. 10, 1913.

GAS POWER DEPARTMENT

Lubrication of Gas Engine Cylinders

By A. L. Brennan, Jr.

The fundamental principle of lubrication is to present an ever-constant film of some non-friction element between two rubbing surfaces irrespective of their speed or the load they carry. Some authorities argue, however, that the cylinder is the all-important part to be considered in an internal combustion motor and allow that most any provision will take care of the remaining parts of the machine. This supposition is alluring as applied to any machine, especially a gas engine, considering that its operation is dependent upon all its parts, not in decreased ratio, according to the apparent value of the part, but entirely. To some this may appear rather unusual but when it comes to practical operation the value of this point is at once recognized. In regard to the cylinder and piston lubrication of a gas engine there are three important points to be considered:

An oil having a flash point of at least 390 deg. F. or nearer 450 deg. is absolutely necessary in order to perform its functions at the high cylinder temperatures.

Another point to be considered is the specific gravity which although given little consideration is important; for unless an oil ranges in specific gravity from 26 to 30 deg. Bé. it will not prove so effective.

The viscosity should range from 180 to 220 at 80 deg. F.

Viscosity is made up of two properties: cohesion and adhesion. The former is that property of the oil that binds its particles together. The greater the cohesion, the greater the pressure which the oil will resist without losing its function of being present in the form of a film; hence an oil possessing excessive cohesion properties will be more suitable under heavy loads, for if the cohesive property were low, even an excessive amount of oil might not prevent the bearing surfaces from rubbing.

Adhesion in an oil is that clinging property which causes it to adhere to other bodies; that is, its tendency to remain on the bearing surfaces. Therefore an oil to be truly effective must be not only cohesive but adhesive as well. The same applies to the non-fluid oils and

All oils to be suitable and effective in gas engine cylinders should be by-products of petroleum, that is some form of mineral oil. Animal and vegetable oils are quite adapted to the other parts of a gas engine where excessive heat is not encountered, but prove absolutely unpractical for the cylinder.

It must be borne in mind, however, that all petroleum oils are not adapted to gas engines in general and can not be adapted to any one machine in particular, and so it is found necessary to select an oil which will meet the several existing contingencies advantageously in order to realize the highest efficiency from a motor. Oils which conform with the specifications mentioned but possess a thin body are undoubtedly the best. Many harbor the erroneous idea that an oil to prove an efficient lubricant must have considerable body, this is altogether wrong as the thinnest oil that will remain in a bearing will invariably give the best results. Of course, under certain circumstances such as in the case of a warm bearing or for heavy pressures oil possessing considerable body will be useful, but otherwise its function will be impaired by the frictional resistance offered by its thick body.

Cylinder oils adaptable to one motor are not necessarily suitable for another. For instance, a high-speed machine lubricated by a splash system, that is, by the cranks dipping into oil contained in the base,—requires a light or medium oil. On the other hand, in heavy duty slow-speed engines dependent upon oil supplied from an oiler of the force feed type, an oil of more body will give slightly better results. But the thickest oil that will flow readily in the grooves, should not be used; instead an oil that will distribute readily and evenly when it reaches the surfaces should be selected.

Thin oil will flow rapidly and in the case of the piston oiling will provide suitable lubrication for the piston rings. This is a most important feature, for if the rings in a gas engine are not kept free from carbon and other deposits they will not move freely in their grooves, inducing additional pressure on the cylinder walls with resultant wear which eventually leads to impaired compression and diminished power.

Faulty cylinder lubrication is invariably attributable to one of two causes, or both—insufficient lubrication with good oil or correct amounts of poor oil. It sometimes occurs, however, that good oil is used in too large a quantity. An excessive amount of oil either of a high or a low flash point will bring about detrimental effects. The first indications in the case of using a poor oil are manifest in decreased power due primarily to the friction offered the surfaces of the piston and cylinder, secondly to faulty compression due to the piston rings becoming inoperative and thirdly to carbon deposits, which materially reduce the efficiency of the valves and impair the function of the spark plug, subsequently inducing pronounced carbon deposits that become incandescent and cause preignition.

Good oil will lead to the same results if too much is used; hence it is evident that a proportional amount of oil, according to the size and speed of a motor, is necessary before the best results can be had. A motor that requires say eight drops of oil per minute per cylinder at 500 r.p.m. will require more oil when the speed is increased, as the piston's travel will cover a greater area. This condition is amply taken care of on motors using the splash feed system or mechanically-driven lubricators, for the supply of oil allowed decreases and increases with the speed; but unless the first adjustment is correct the increased amounts, when the motor is speeded up, will be proportionately at fault.

By procuring the best grades of any of the popular brands of cylinder oils an operator may feel confident as to the thorough lubrication of the piston directly and indirectly. By "indirectly" reference is made to the wrist pin. This important yet often neglected bearing is in many instances taken care of by a portion of the oil supplied to the piston surfaces. This oil works along the surfaces of the piston as far as the oil retaining ring which has a tendency to maintain a constant feed of oil, but it at the same time directs a small amount of oil to the grooves leading to the wrist pin. Here again the importance of employing an oil of comparatively thin body yet of good viscosity is essential for the creeping ability of a thick oil might militate against its travel to such a degree as to render it impractical for this purpose. Also an oil of low flash point would quickly carbonize in the high temperatures and the rings.

Operators have a tendency to put up with an inferior grade of cylinder oil rather than pay the high prices now asked for a first-class product. This is a poor policy for not only is the output of the machine reduced but the wear and tear to which a motor is subjected when employing inferior grades of oil is decidedly on the negative side of a profit and loss account. Poor lubricating oil not only impairs the mixture by a partial combustion of its own elements but by not keeping the wear at a minimum, allows lower compression which in turn puts greater demands upon the ignition current. Considering the mechanical depreciation of a motor alone the slight additional expense of using the very best oils obtainable is to be recommended.

A good plan to follow, when considering the high price of the best oils, is to use the best grade cylinder oil for the cylinder only and any good machine oil for the other parts of the motor. Of course, in motors employing the splash feed system, there will be but little saved, as nearly all the bearing parts are looked after by the "splash," but on motors having force lubricators or oil cups for the cylinders a marked saving can be attained by using two oils. Find out by experimenting as early as possible the correct amount of oil necessary to produce the best results and then keep that amount constant.

If by any chance you are compelled to employ another make or grade of oil which may appear to the naked eye to be similar to the first, you should use the same precaution to ascertain if your surmise is accurate, that you did at first, for the new oil may not feed as fast which would result in under-lubrication or it might be even more subject to heat than the first and feed at a greatly increased rate with the detrimental effects pertaining thereto.

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A natural-gas pipe line, 124 miles long and 12 in. in diameter, is now in course of construction in California for the Midway Gas Co. The operating pressure on this line will be 450 lb. per sq.in., the highest yet employed for this purpose.

The Junkers Engine—VII

By F. E. Junge

The increase in dimensions of small engines in direct ratio to the growth of capacity is restricted in several ways. Good scavenging and perfect combustion exact a ratio of length of stroke to cylinder diameter of at least 1.5. The piston speed also acts prohibitively, inasmuch as it depends upon the inertia pressures, the frictional resistances, and the lubricating conditions. With the enlargement along these lines limits of practicability are soon reached, making reliability and economy a serious problem. The following power schedule, of two-stroke-cycle, single-acting Diesel and Junkers engines in tandem arrangement, will elucidate the foregoing.

The following assumptions are made: in the Diesel engine a ratio of stroke to diameter of 1.5, and a piston speed of 16.4 ft. per sec. (5 m. per sec); in the Junkers engine a ratio of stroke to diameter of 2, and a piston speed of 11.5 ft. per sec. (3.5 m. per sec.). The mean pressure referred to the shaft horsepower has been assumed at 71.2 lb. per sq.in. (5 kg. per sq.cm.) for both engines. For the two-stroke-cycle single-acting Diesel engine the indicated horsepower per cylinder is then expressed by the equation:

$$I.hp. = 0.167 \frac{\pi d^2}{4}$$

For the Junkers tandem engine

$$I.hp. = 0.467 \frac{\pi d^2}{4}$$

From these formulas the following table has been computed:

Diesel engine with piston speed = 5 m. per sec.					Junkers tandem engine with piston speed = 3.5 m. per sec.					
Output in hp.	Diameter of cylinder, mm.	Stroke, mm.	R.p.m.	Maximum piston-force for p = 40 kg. per sq.cm.	Length 7 x stroke, m.	Diameter of cylinder, mm.	Stroke, mm.	R.p.m.	Maximum piston-force, kg.	Length 8.5 x stroke, m.
300	480	720	210	72,000	5.0	290	580	360	26,p00	4.9
400	550	825	180	95,000	5.8	330	660	320	34,000	5.6
600	680	1020	147	145,000	7.2	400	800	260	50,000	6.8
800	780	1170	128	191,000	8.2	470	940	224	69,000	8.0
1000	880	1320	114	243,000	9.2	520	1040	200	85,000	8.9
1200	960	1440	104	290,000	10.1	570	1140	185	102,000	9.7
1500	1070	1600	94	360,000	11.2	640	1280	184	129,000	10.0
1800	1170	1760	85	430,000	12.3	700	1400	150	154,000	11.9

This tabulation shows the superiority of the Junkers engine in tandem arrangement over the two-stroke-cycle, single-acting Diesel engine. Although the ratio of stroke to diameter has been taken very large, the total length of the Junkers engine is less than that of a Diesel engine of the same power. Under the same circumstances the maximum gas pressure on the piston is approximately three times less than that in the Diesel engine. The weights and first costs are decreased accordingly.

Regarding the development of the power-transmitting mechanism of the Junkers engines as a combination of the mechanism of several single-acting Diesel engines, there is this to say: In a two-cylinder two-stroke-cycle vertical Diesel engine (see Fig. 37) the energy is transmitted to two cranks by two connecting-rods. On the cylinder cover the scavenging valves are fixed. Along the circumference of the cylinder walls, ports are arranged for the exit of the burnt gases, together with the scavenging air which expells them. These slots are placed near the outer dead center position of the piston. Three main bearings are necessary. While an expansion takes place in the one cylinder, compression is accomplished

in the other, and *vice versa*. In the upper dead center position the heat supply or fuel injection takes place. In the outer or lower dead center position of the piston the products of combustion are expelled. For every revolution two power strokes are obtained.

Through the action of forces in the cylinders during both the working and compression strokes, a tension is created in the engine frame of varying amount for each cylinder support. In the transmission of the piston force of the one cylinder on the crankshaft during the expansion stroke a vertical downward reaction is set up in the two corresponding main bearings. The same happens in the force transmission from the crankshaft to the piston of the other cylinder during the compression stroke. The inertia forces due to the reciprocating parts augment these loads. Thus during every revolution the three main bearings have to support loads varying in amount.

The strongly fluctuating load to which the foundation is exposed during every revolution excites vibrations in the latter. Special stress should be laid on the large amount of energy efflux necessary for the maintenance of these vibrations. This energy is, of course, lost and detracts from the effective output.

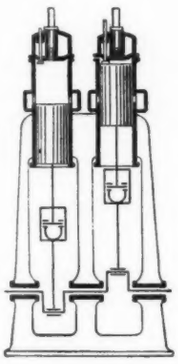


FIG. 37

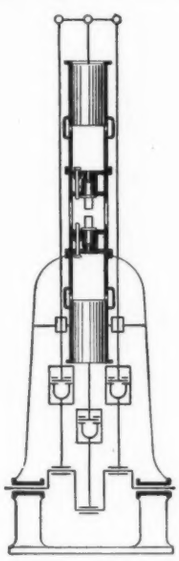


FIG. 38

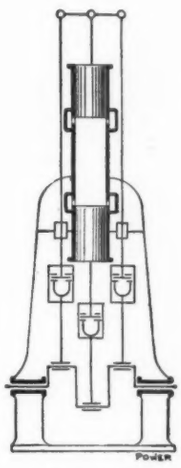


FIG. 39

To attain favorable dynamic conditions the following changes may be resorted to: Two cylinders may be set one above the other, and the transmission of forces arranged as shown in Fig. 38. Then by the medium of three connecting-rods the mechanical energy is transmitted to three cranks. In addition, one traverse piece and two side rods are necessary. The scavenging valves are arranged in the cylinder covers, as in Fig. 37, and the exhaust ports are placed along the circumference of the cylinder, in the outer dead center position. The expansion and compression strokes take place simultaneously in both cylinders, and at the extreme inside position of the pistons, fuel injection takes place, while the burnt gases are expelled in the outer dead center position. For every revolution there is one power stroke as against two in Fig. 37. Assuming both types of engines to have equal diameters of cylinders, equal lengths of stroke and the same number of revolutions, their outputs will be equal.

Provided the forces acting on the cylinder covers are the same they neutralize each other completely. The forces in the engine which severely tax the frame in Fig. 37 are obviated, the acting forces being restricted to the power-transmitting mechanism. As the lower piston is connected to the middle crank its force is divided equally between the two main bearings. The direction of these loads is vertically downward and upward, alternately; the same being true for the two side cranks. Only the forces encountered here are always in opposition tc those just mentioned. The main bearings are thus disburdened of any loads that might otherwise exist. A small residual load, however, is retained. For this the obliquity of the connecting-rod is responsible. In the arrangement illustrated in Fig. 38 the coefficient for the fluctuation of speed is twice as large as in the case of Fig. 37. For the same uniformity of twisting moment the former will require a heavier flywheel. But as resulting advantages of the second arrangement, the main bearings are relieved of loads, the forces are balanced in the mechanism instead of the frame, and the inertia is balanced in coincident lines.

Dispensing with the cylinder covers and replacing the two cylinders by a single one, the type shown in Fig. 39 is arrived at. Compared with the type in Fig. 37, this third type possesses the same advantages as the second (Fig. 38). In addition, however, there are the following favorable features: the cylinder covers are obviated and the engine becomes shorter. The scavenging valves are dispensed with, as the row of ports in the cylinder wall are utilized for scavenging purposes, and the scavenging conditions are made more favorable. Finally, the harmful heat dissipating surfaces at dead center are restricted to a minimum. This third type corresponds to the Junkers engine.

CORRESPONDENCE

What Causes the Pound?

We have a three-cylinder 80-hp. Sheffield oil engine direct connected to a 50-kw. alternator. The fuel is 40 deg. distillate, although gasoline is used for starting. Ignition current is supplied from a 110-volt exciter circuit with three 37-volt lamps in series and three groups in parallel (nine lamps) to each cylinder; this furnishes a current of 14½ amp. to the plug.

A pound has developed in one cylinder, occurring at the instant of ignition. Advancing or retarding the spark makes no difference; nor does injecting more water with the fuel. We have tried preheating the fuel, but this does no good. The pound can be stopped by using a richer mixture which, however, reduces the power of the engine. The engine does not pound on light loads.

Can any reader enlighten me as to the probable cause of the pound?

GEORGE E. ABEL.

Ness City, Kan.

To remove enamel from almost any metal, according to "Revue Produits Chimiques," it is necessary only to heat the enameled object in a furnace or muffle to a red heat and cover the surface with powdered potassium cyanide. After cooling, the enamel will be soluble in water.

REFRIGERATION DEPARTMENT

A Gas Engine Refrigerating Plant

Before the sixth annual meeting of the American Society of Refrigerating Engineers in 1910, R. H. Tait described a refrigerating plant in which a horizontal, two-cylinder, tandem, double-acting, 14x21-in. ammonia compressor, running at 128 r.p.m., was direct driven by a two-cylinder, tandem, double-acting, 21x30-in. gas engine operating on producer gas made from Illinois bituminous coal. The gas producer is of the suction-updraft type, about 10 ft. 2 in. in diameter and 12 ft. high, with 56.75 sq.ft. of grate area.

The general arrangement of the installation is shown in Fig. 1, where *A* is the producer, *B* the uptake open at the bottom to a water-sealed compartment of the hydraulic main *D*, *C* the tower scrubber, containing six water sprays and open to the hydraulic main from which the water is continually overflowing through an 18-in. seal. The water is supplied through the tower scrubber and through four large sprays in the hydraulic main proper. The tar extractor *E*, running obout 1350 r.p.m., is direct connected to a 15-hp. motor, while the gas blower *F* is connected to a 10-hp. motor and runs at a speed ranging from 800 to 1600 r.p.m. The gas blower is bypassed into the hydraulic main, which is sealed so as to deliver the required pressure at the engine. The gas main is a 10-in. pipe having a fall toward the blower.

The uptake *B* is a 16-in. vertical pipe resting on a 10-in. pipe open to the water-sealed compartment of the hydraulic main, allowing the deposit of soot to fall into this compartment. This offset from 16 in. to 10 in. provided a shelf on which the soot built up, and the opening through the 16-in. pipe was reduced to the size of the 10-in. pipe, upon which it rested. The horizontal connection between the uptake *B* and the tower scrubber *C*, also filled up, effectually preventing any draft through the producer and shutting off the gas supply to the engine. Upon opening the uptake the gas in the pipe took fire, the soot burned out, the charred mass was removed and to prevent future accumulations, scrapers were inserted which could be used during operation.

The tower scrubber with its six horizontal sprays partially cleans and washes the gas before it enters the hydraulic main, which is a horizontal tank, 4 ft. in diameter, 20 ft. long and made of ⅜-in. iron. The function of the hydraulic main is to reduce the velocity of the gas, thereby allowing impurities to precipitate which are held in suspension at higher velocities. The water supplied is sufficient to carry the greater part of these impurities to the sewer, and the tar accumulation in the bottom is removed about once in two weeks.

At the outset the producer was started on gas-house coke and the fire allowed to build up gradually to heat up the lining. After the heavy firebrick lining was hot, with a proper depth of fuel bed, bituminous coal was fired. It developed that bituminous coal, with the same handling as coke, clinkered and the bed became unmanageable. It was then decided to build up the fuel bed with buckwheat anthracite, then gradually introduce the bituminous coal. Although somewhat more successful, the experiment proved a failure on account of the excessive expenditure of labor necessary to keep the fuel bed in condition to make gas.

It may be well to state here that coke, being light, requires frequent poking down to close the voids or blow-holes in the bed. Anthracite coal tends to pack, offering a more uniform resistance to the draft. This bituminous coal, immediately after charging, is easily spread and acts similar to anthracite, but when once hot it exhibits a marked tendency to run together and clinker if poked.

Consequently the excessive poking was eliminated and the fuel was spread immediately after charging. The fuel bed, however, would burn through at the lining, forming blow-holes. To overcome this difficulty without resorting to poking, ash was allowed to accumulate at the lining, forming a dish-shaped ash bed, varying from 3 to 6 in.

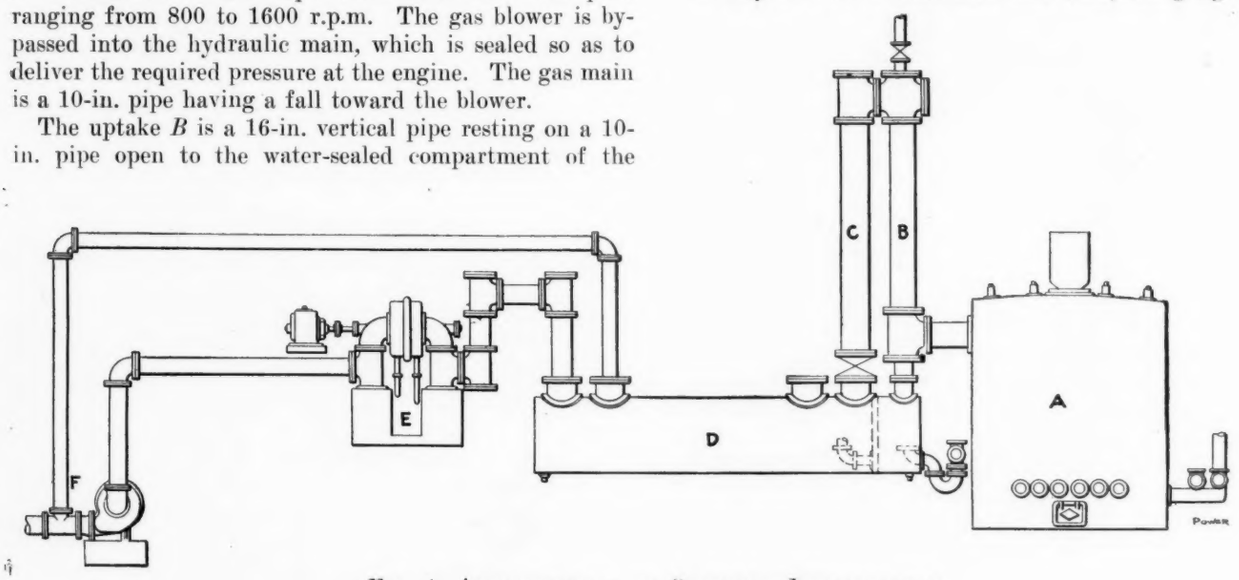


FIG. 1. ARRANGEMENT OF ORIGINAL INSTALLATION

in the center to from 6 to 12 in. at the outer edge. Excessive accumulation of ash was dropped into the ashpit by using short pokers through the side poke holes.

It developed that the deeper the fuel bed the more uniform the quality of the gas. Different depths of bed were tried, ranging from 2 to 6 ft. With the 6-ft. depth of bed, the top of the coal in the center of the producer was above the bottom edge of the petticoat and the coking around this bottom edge prevented the proper spreading of the fuel; hence the bottom edge of the petticoat determined the maximum depth of the fuel bed as 4 ft. The operation resolved itself into charging coal at certain intervals, depending upon the load, spreading the charge immediately after dumping and cleaning out the ash about every eight hours.

Instead of employing a gas tank to furnish a constant supply of gas to the engine, the speed of the blower was varied with the changes in the load and the gas was delivered to the engine at a pressure of about one inch.

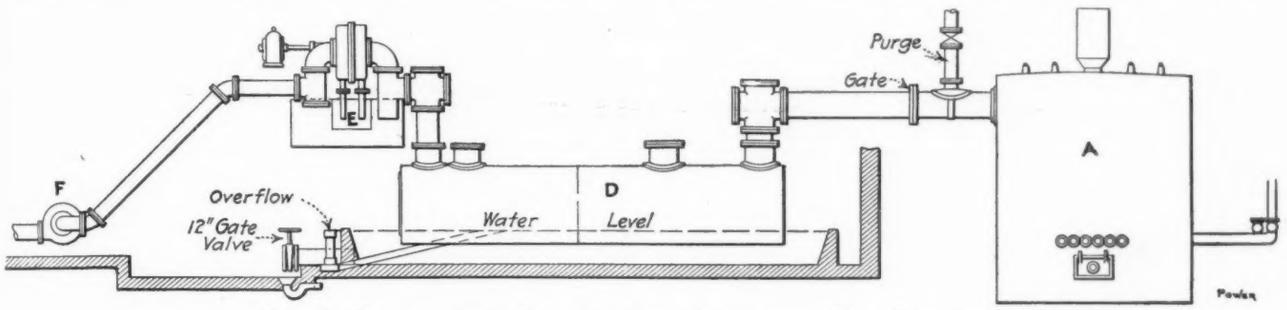


FIG. 2. PLANT AFTER GAS-CLEANING APPARATUS WAS REMODELED

Since this paper was presented some changes have been made in the gas-cleaning apparatus to improve operating conditions. These changes were enumerated and the results of tests on the plant were given by E. W. Gallenkamp, Jr., in a paper read before the society at the December meeting. Fig. 2 shows the remodeled plant. The uptake *B* and the tower scrubber have been eliminated and the gas passes directly into the hydraulic main *D*. The tar extractor was placed close to the hydraulic main and the bypass around the blower is now used only in starting the engine. The hydraulic main was cut off at the bottom one-third of the way up, partitioned in the middle and the entire tank water sealed. The partition has a 6-in. hole through which the gas must pass. A flat cast-iron ring about 6 ft. inside and 8 ft. outside diameter was placed upon the grate bars in the gas producer to obviate the formation of holes in the fuel bed at the sides. Scrapers were inserted in the connections between the producer and the hydraulic main.

To obtain the best results with the producer, the coal is spread immediately after charging and must not be disturbed after it becomes hot. If a coke crust overspreads the surface of the bed, preventing a free passage of gas, this crust should be broken up, inserting the poker to not more than 12 in. below the top of the bed. The ashes are drawn once in three hours and every two weeks the machine is shut down for three or four hours to clean the gas-cleaning apparatus and the igniters on the engine.

After the gas leaves the tar extractor it still carries tar in suspension, but is completely purged of soot. The governor valves on the engine show a heavy coating of tar at the end of two weeks, and, in fact, wherever the gas changes its direction or velocity tar is deposited. The tar extractor is the main source of trouble, as a new rotor is required about every five months.

Originally the governor on the engine was connected up for a uniform quality of gas, varying only with the quantity of mixture. The mixture of air and gas was to be regulated by hand. The variable gas quality made it necessary to alter this regulation, so that now the governor varies the mixture. In addition the sensitiveness of the springs in the governor was diminished to reduce the speed variation. When the governor rides near the top, the speed of the engine is about 132 r.p.m., and when riding near the bottom, but still within its range, the speed is about 120 r.p.m., giving an average of about 127 r.p.m. By means of a turnbuckle in the rod from the governor arm to the governor valve shaft, the speed of the engine may be reduced to 80 r.p.m., a reduction of 47 r.p.m. This reduction in speed being effected by wire drawing of the gas increases the tar deposit and necessitates more frequent cleaning of the valves. Before the application of the turnbuckle on the governor rod, operators frequently reduced the speed of the engine to 30 r.p.m. by throttling at the main gas valve, which resulted in completely filling the 62-in. pipe just in front of the valve with tar.

TEST OF PLANT

The manufacturer of the gas engine and producer guaranteed under normal load a brake horsepower-hour on not more than 1½ lb. of coal of the same grade as that used under the boilers. To demonstrate this, indicator diagrams were taken and the horsepower calculated from the diagrams. After a week's run the engine had developed on an average 318⅓ i.hp., while the compressor indicated horsepower was 222.05, representing a combined mechanical efficiency of 69.7 per cent. It is assumed that the mechanical efficiency is equally divided—that is, 84.85 per cent. for each machine, making about 170 b.hp. for the engine, which is about 75 per cent. normal load. The results show an average consumption of only 1.54 lb. per b.hp.-hr. of washed Illinois bituminous coal, averaging 11,028 B.t.u. per lb.

Starting on Nov. 30, 1911, a six-day test was conducted to determine the number of pounds of ice per ton of coal it was possible to produce under normal operating conditions, which were not ideal by any means. Readings were taken every three hours and the results plotted in charts were presented in the paper.

The condenser pressure was carried at 180-lb. gage and the suction pressure at 17 lb. The temperature of the atmosphere averaged 37 deg. F., the brine in the freezing tanks 15 deg. and the water entering the forecooler 53 deg., where it was cooled to about 35 deg. Out of a nor-

mal capacity of 816 cakes, averaging 405.8 lb., per day, about 600 cakes were pulled daily during the test, except on Sunday, when 352 cakes were pulled in half a day. The total amount was 3425 cakes, amounting to 115.8 tons per day of 24 hr.

In the 144 hr. of the test there were three stops, one for 15 min. and two for 20 min., a total of 55 min., leaving 8585 min. of actual running time. The total number of revolutions as shown by a recorder was 993,076, or an average of 115.68 r.p.m. The cubic-inch displacement per revolution of the compressor is 12,930.96 and per minute would be 1,495,853.45 cu.in.

This means 12,918 cu.in. displacement per minute for a ton of ice per day. The ice tank contains 1632 cans, which makes the rate of freezing per cake about 68⅔ hr. The results of the test are partially summarized in Table 1.

TABLE 1. SUMMARY OF TEST

Duration of test, hr	144
Total amount of ice pulled, cakes	3,425
Av. wt. of cakes, lb	405.8
Total weight of ice pulled, tons	694.9
Average amount of ice per day, tons	115.8
Total coal consumption, lb	62,900.0
Total coal consumption, tons	31.45
Av. coal consumption per day, tons	5.24
Total ashes, lb	6,650.0
Average ashes per day, lb	1,108.4
Ash in coal, per cent	10.6
Ice per ton of coal, as fired, tons	22.10

The power consumed by auxiliaries is estimated as accurately as possible in Table 2.

TABLE 2. POWER CONSUMED BY AUXILIARIES

	Horsepower
Driving ice-tank agitators	21.24
Driving tar extractor and igniter generator	17.41
Circulating jacket cooling water	3.13
Circulating oil in lubricating system	0.25
Pumping water for condenser and cooling coils	9.51
Pumping distilled water	1.15
Furnishing compressed air for hoisting and starting	5.75
Total	58.44

Assuming 1.81 lb. of coal per horsepower-hour, including an allowance for transmission from engine to motor, would give a coal consumption for auxiliaries of

$$58.44 \times 1.81 = 2540 \textit{ lb.} = 1.275 \textit{ tons per day}$$

Including auxiliaries, the total coal consumption on the above assumptions would be 5.25 tons plus 1.27 tons, or 6.52 tons per day. The ice per ton of coal as fired would then be:

$$115.8 \div 6.52 = 17.8 \textit{ tons}$$

Summer conditions would call for more water on the ammonia condenser, as the temperature of the water at that season is often as high as 88 deg. F., and the temperature of distilled water to the forecooler as high as 96 deg. F. The temperature of the atmosphere would average for the hot months about 79 deg.

The cooling water from the ammonia condenser is used in the gas-cleaning apparatus and is ample for the purpose, but the tar and impurities taken up make the water unfit for use in a cooling tower. Where water is obtained under difficulties and at a considerable cost, a plant of this arrangement would be prohibitive. With the exception of the tar extractor, the installation has given good service throughout the summer. The shutdowns, as shown on the record for 183 days, from Apr. 1 to Oct. 1, 1912, are given in Table 3.

TABLE 3. SHUTDOWNS IN 183 DAYS

	Hours	Minutes
Due to tar extractor	51	25
Fourteen Sundays	49	..
Due to producer, engine or compressor	32	5
Total	132	30

❋

Physical Properties of Ammonia

In a paper read before the American Society of Refrigerating Engineers, Prof. Lionel S. Marks and F. W. Loomis, of Harvard University, gave the results of the most important experimental work published so far, some analysis of those results on thermodynamic and other grounds, and a critical discussion of the tables that have been prepared by various authorities.

In conclusion it was agreed that the best table of the properties of saturated ammonia is that given by Wobsa in his second paper. This table transformed into English units, is given herewith, but should not be confused with the earlier table by the same authority. The present state of knowledge of the properties of ammonia is much more satisfactory than it was ten years ago, and the accompanying table may be used for ordinary engineering calculations with some confidence.

For greater accuracy, it is necessary that the specific heat of the liquid ammonia should be further investigated and that confirmatory data should be obtained on specific volumes of the saturated and superheated vapor. The vapor pressure needs some further determinations at low temperatures, but it is probably known with sufficient accuracy throughout the range used in engineering practice.

In another paper on the same subject, William E. Mosher presented an interesting discussion on former experimental data and proposed several formulas to express the various properties of ammonia. The values resulting from the use of these formulas agree closely with experimental data and by Mr. Mosher are considered more accurate than the values given by Wobsa, although the difference between the two sets of data is not very great. No working tables of the properties of ammonia were included in the paper, but such tables have been prepared for both the saturated and superheated vapor and they will appear shortly, together with a Mollier diagram for ammonia, in a bulletin of the engineering experiment station of the University of Illinois.

TABLE OF THE PROPERTIES OF SATURATED AMMONIA VAPOR, BASED ON WOBSA'S SECOND EQUATION (1908).

Temperature Deg. F.	Pressure, lb. per sq.in.	Specific Volume, cu.ft. per lb.	Density, lb. per cu.ft.	Heat of the Liquid B.t.u.	Total Heat, B.t.u.	Latent Heat. B.t.u. Total	Internal	External	Entropy. Liquid n.	Vapor N.	Difference $N-n$.
—22	16.98	15.80	0.06329	—58.91	524.9	583.7	534.2	49.50	—0.1265	1.2090	1.3355
—13	21.58	12.60	0.07936	—49.32	528.1	577.4	527.2	50.18	—0.1048	1.1891	1.2939
—4	27.14	10.13	0.09871	—39.65	531.0	570.6	519.8	50.78	—0.0835	1.1699	1.2534
+5	33.81	8.323	0.1201	—29.90	533.9	563.7	512.3	51.34	—0.0622	1.1524	1.2146
14	41.69	6.744	0.1483	—20.04	536.2	556.2	504.4	51.84	—0.0414	1.1351	1.1765
23	50.95	5.569	0.1796	—10.06	538.6	548.6	496.4	52.23	—0.0206	1.1174	1.1370
32	61.71	4.635	0.2157	0.	540.7	540.7	488.0	52.63	0.	1.1011	1.1011
41	74.21	3.885	0.2574	10.17	542.5	532.5	479.5	52.97	0.0205	1.0852	1.0647
50	88.55	3.275	0.3053	20.45	544.0	523.6	470.3	53.22	0.0405	1.0690	1.0285
59	104.90	2.776	0.3602	30.83	545.2	514.5	461.2	53.39	0.0607	1.0540	0.9933
68	123.5	2.366	0.4227	41.31	546.3	505.1	451.6	53.50	0.0805	1.0392	0.9587
77	144.4	2.027	0.4933	51.91	547.0	495.2	441.7	53.50	0.1003	1.0245	0.9232
86	168.1	1.745	0.5730	62.55	547.4	484.9	431.4	53.42	0.1197	1.0102	0.8905
95	194.5	1.509	0.6627	73.42	547.7	474.3	421.0	53.33	0.1392	0.9963	0.8571
104	224.0	1.309	0.7640	84.37	547.7	463.3	410.2	53.10	0.1584	0.9826	0.8242

EDITORIALS

Surface Condensation

It is always a bit difficult to grasp the significance of any discussion relating to molecules. We cannot see nor feel a single molecule and, if we hadn't more faith than the farmer at the circus who would not believe there was such an animal even when he saw the giraffe, we would refuse to believe that anything so small as a molecule can exist. To comprehend the actions of these infinitesimal particles it is helpful to consider them magnified to the size of some familiar object as a billiard ball or a ball of putty, as Professor Munro has done in his analogy for initial condensation on page 128.

For simplicity's sake he has taken other license with the molecules than simply to enlarge them, giving them difference of hardness, whereas we have been taught to believe that the simon pure molecule is unyielding and incompressible. He has also overlooked the fact that when the steam molecules become billiard-ball size the iron molecules of the cylinder walls do also and the surface impinged against is not smooth and continuous.

Instead of impact and rebound of the steam molecules taking place, it is supposed in reality that these molecules have a planetary motion around the molecules nearest the surface of the iron and then return on their course, or remain in the same neighborhood with the iron molecules. The latter is the phenomenon of initial condensation. The steam molecules lose sufficient velocity (or heat) to the cooler and much less active iron molecules in raising the velocity (or heat) of the latter for the steam to condense to water.

Although the hypercritical may discover these weak spots in Professor Munro's analogy, we believe he was justified in taking the liberties he did and rather like his simple explanation of initial condensation.

❖

What Is a Horsepower?

A horsepower equals 33,000 foot-pounds per minute. or 550 foot-pounds per second.

What is a foot-pound?

The amount of energy necessary to overcome a pound of resistance through a foot of space, as when a pound weight is lifted one foot, or a pressure of one pound is exerted upon a body (as, for instance, a piston) to move it through one foot.

What is a foot?

By an Act of Congress, July 28, 1866, the meter, which contrary to the general impression, is the fundamental standard of length in the United States, was declared to be equivalent to 39.37 in. The legal foot in the United States is therefore 0.304801 of a meter. The international meter is the length of a bar of platinum and iridium preserved at the International Bureau of Weights and Measures near Paris. The United States owns two duplicates of this bar.

There is, therefore, a standard fixed value for the foot, so far as the United States is concerned.

What is a pound?

The pound is defined in terms of the kilogram as the foot is defined in terms of the meter. For purposes of discussion we may consider that there is at Washington a standard piece of metal, the weight of which is one pound.

What do we mean by weight? Ah, there's the rub.

Weight is the measure of the force of gravity upon a body. If this standard pound weight is put into one of the pans of a pair of balances and enough of some other material is put into the other pan to balance it, the amount of matter so measured is a pound, because the earth pulls upon it with the same force that it does upon the standard pound weight.

But the pull of the earth, the attraction of gravitation, is different at different parts of the earth. At the equator it is partly neutralized by the centrifugal force, due to the earth's rotation upon its axis, and the greater distance from the earth's center of gravity. The same thing applies to elevation above the sea level, so that the force with which the earth attracts this pound weight varies with latitude and elevation.

If the scale with the standard pound weight in one pan and the mass of some other material which will balance it is taken to the poles or to the equator it will still balance, because the lesser or greater force exerted by the earth will be exerted upon the two bodies alike. Determined in this way "a pound is a pound the world around," a mass equal to that of the standard pound weight.

Suppose, however, the standard pound weight to be hung upon a spring scale at the latitude of Paris, and a mark to be made opposite the pointer indicating one pound. If this spring scale, so weighted, be taken to the equator the pointer will stand above the mark which was made at Paris, on account of the lesser attraction of the earth upon the weight, and it would take a greater amount of matter to indicate a pound upon this kind of a scale at the equator than further north, if the matter upon which the earth will exert the force necessary to stretch the spring until the pointer reaches the mark made in Paris be accepted as the pound.

Take a mass of 55 pounds. It would balance 55 of the standard weights anywhere in the world, but it would take a greater force to lift it at Berlin or London than it would at Havana or New Orleans. If it is lifted ten feet in a second, work is done at the rate of one horsepower according to the above definition, but there has not been so much energy used at the southern as at the northern latitudes.

Is the horsepower then an indefinite variable quantity, varying with latitude and elevation?

The U. S. Bureau of Standards says that it is, and recommends that it be given stability by being considered as equivalent to 746 watts, which would be 550 foot-pounds per second at about the latitude of London and sea level.

It is an absolute unit only if there is as absolute a

definition of the pound as there is of the foot; and if the pound is a measure of force as well as of mass.

There is such an absolute definition in common use. It is the force with which gravity attracts the standard pound weight at 45 degrees latitude and at sea level, the force which acting constantly will accelerate that pound of mass at the rate of 32.174 feet per second.

Having thus an absolute unit of force and of space the absolute value of the foot-pound is fixed, and 550 of these foot-pounds per second is just as absolute a value for a unit of power as the watt.

The values of the horsepower and the watt are thus absolutely fixed and the horsepower is equal to 745.7 watts. The only way to make it equivalent to 746 watts without changing the value of the watt is to change the value of the horsepower. This involves either using 550.22+ foot-pounds per second, 33,013.2+ foot-pounds per minute as the horsepower or changing the accepted value of the pound. Nobody would countenance the changing of the horsepower to this inconvenient and incomplete number of foot-pounds, and the changing of the value of the pound is too serious a matter to consider for the purpose simply of being able to express horsepowers in even figures in terms of watts.

Unless this is done the best we can do is to recognize that a horsepower is 745.7 watts and use the 746 as the nearest round number as we do 778 as the mechanical equivalent of the heat unit in calculations which do not involve the greater degree of accuracy. This is the value given in the conversion table accompanying the Marks-Davis steam tables, and that arrived at in Mr. Kent's consideration of the subject on page 109 of this issue.

The Commercial Engineer

Representatives of water, electric-light and other public-service corporations, in defense of their rates invariably put stress upon the fact that it is service they are selling. They must maintain a great organization of skilled specialists and a great aggregation of plant and apparatus so that when one turns the cock water will come, or when one wishes light that there shall be light.

Carrying the thought a little further, when a man sells an engine he does not sell merely so much steel and cast iron; nor merely the combination of those materials with the work of the foundryman and the machinist, who have fashioned and fitted it. If the salesman is a commercial engineer in the highest significance of the term he sells a solution of his customer's trouble and a fulfillment of the needs and purposes in his particular line. He is able to do this because he is a specialist in that line. He knows what is available; what its capacity, capabilities and limitations are, and if he is disinterested and knows the engineering end of his line as well as he does his discount list and the commission rates, he is the best adviser that an intending purchaser can possibly have.

But when one holds a brief for a particular line, and when one's income depends on the amount of his current sales, it is not easy to be disinterested.

Here is a textile concern, say a carpet mill, that can use up all the exhaust steam it can make. The manager writes to a builder of condensers that he is in the market for a large order. Will that condenser salesman go there and tell that man that he has no need for a condenser, or will he try to close him for the biggest order that he can land? Will he approach that job as an engineer or as a condenser salesman?

Is his success measured by the soundness of the engineering advice that he gives (for nothing when it does not lead to business) or by the dollar's worth of orders that he sends in?

The concern which adopts the altruistic attitude will acquire a great standing in the course of time. If the carpet maker ever did need a condenser or had anything to say about the buying of one there is only one place to which he would naturally look and one man to whom he would first turn for advice; but how far, from a business and commercial point of view, can a concern go in the direction of maintaining a corps of specially trained men to travel at its expense and advise people not to buy its goods?

There is no doubt that the manufacturer is often imposed upon with respect to engineering service and advice. Recently bids were invited for an installation in which each bidder was required to submit plans and specifications of his outfit, and when they were all in, the consulting engineer of the purchaser simply made up from the symposium an excellent layout embodying the best suggestions of all the contestants and had it built himself.

But if the selling representative is to be simply a salesman, why the term "commercial engineer?"

The title is suggestive of an intelligent effort to apply real engineering to the disposition and application of engineering devices and appliances, and to remove from commercialism, as applied to engineers, some of the opprobriousness which has resulted from abuses committed in its name.

It is for those who have adopted the title commercial engineer to emphasize the newer and better meaning of the term; to demonstrate that in the selling of engineering apparatus and materials, real engineering knowledge of the merits, capabilities, yes, and limitations of one's wares and ability to meet the engineer users of those wares upon a common plane of mutual engineering interest are worth more than lung power or eloquence.

These Lawyers!

For real comedy what can equal the outcome when a smart lawyer tries to heckle the expert witness and gets the worst of it—as usual? Up in New England a few days ago, as related in the letter on page 128 entitled "Engineers' License Law Violation," the defendants' lawyer could not understand why a first-class licensed engineer should be worth forty dollars a week. To him an engineer is a part of the great dinner pail and overall fraternity; he does not sense it that knowledge and judgment are as essential to the successful handling of a power plant as to anything with which he has to do. Nor could he appreciate why one plant may require a higher salaried man than any other. It is the old, old story of the futility of trying to make the legally-trained mind grasp the most elementary principles of other professions. This is no reflection on lawyers as a class, for they should not be and are not expected to know everything about all vocations. The pity of it is that they feel that their success depends on their assuming the all-wise air; otherwise they would save themselves many embarrassing predicaments.

READERS WITH SOMETHING TO SAY

Operating Costs of Hydraulic vs. Electric Elevators

It is not often that the operating engineer has an opportunity to determine or get acquainted with the cost of operating an electric and a hydraulic elevator for the same building and kind of work. I was fortunate enough at one time to operate a plant where electric elevators were substituted for the hydraulic ones then in service. This plant consisted of five 72-in. horizontal return-tubular boilers and twelve electro-magnet-control hydraulic elevators, running 450 ft. per min. The building was a 12-story office structure. This plant supplied steam and hydraulic power to three office buildings of this kind.

The pressure tank was badly pitted and later condemned by the inspector. I recommended the purchase of a new pressure tank, as this was the only part of the equipment that was crippled. Other influence was brought to bear, however, and the management decided to take out the hydraulic elevators and install electric machines. This called for an expenditure of over $13,000.

These buildings were using central-station current when the hydraulic elevators were in service. The exhaust steam from the elevator pumps was sufficient to heat the building, giving a back pressure of nearly 2 lb. in the coldest weather. Live steam was very seldom used.

I kept records of the coal consumption for one year and compared those covering the same summer months, while the hydraulic elevators were used, to those covering the same months when the electric elevators were in service. I found that the coal consumption had decreased 24 tons per month over the year previous, with other conditions practically the same. It must be remembered that no steam was being used for heating purposes during these months and that the decreased coal consumption of 24 tons represents quite accurately the saving made by the electric elevators. But as soon as the buildings had to be heated, instead of using exhaust steam as in the year previous, it was necessary to use live steam at about 2-lb. pressure, through a reducing valve.

Records of the coal consumption for the winter months show that 20 tons more per month were required when the electric elevators were used over that necessary when the hydraulic elevators were in service. This average increase of 20 tons per month continued year after year. It must be remembered that the cost of the current for the electric elevators must still be added to the coal bill, because during previous years this coal consumption was that incurred by running the elevators and furnishing the buildings with heat, whereas with the electric elevator the coal consumption increased and still the cost of the current had to be added to the coal bill. I have records of the kilowatt-hour consumption, but as the reading includes lighting and motor current, all on the same meter, the current used by the elevators could not accurately be estimated.

Had it not been that the space formerly occupied by the hydraulic elevators had been rented there would have been a considerable loss, due to the change in elevator service.

On the whole, electric elevators are to be preferred over hydraulic ones, because there are very few accidents that can be traced to this type of machine. They are quickly stopped and started, and never safer than when stopped, for when the lever is central or the main switch is pulled, the brake spring grips the brake and will not allow any movement of the car. This is entirely different with the hydraulic types, as they are never more unsafe than when they are stopped; one cannot tell at what instant they will start to creep up or down when the lever is central. This is due to leakage of the pilot valve or the cup leathers in the main valve, which allows water under pressure to be admitted to the plunger cylinder and thereby start the car. Numerous accidents have been caused by this creeping, as operators frequently leave the car, and when they return after a few minutes' absence they find it creeping up and try to get aboard it, and in so doing are caught between the car and the floor.

Pilot valves often become plugged in the ports with packing fiber, and this trouble makes it impossible to hold the car in one position for more than a minute or so. If the pilot and main valves are in perfect condition the car will stand for any length of time if the lever is central; but on account of the continual wear they are subject to, it is seldom that one finds these valves to be perfectly tight.

There are a number of electric elevators in this city, but I cannot recall one accident to them or to their operators. It would be interesting to read statistical reports of accidents to hydraulic and electric elevators respectively.

A. C. WALDRON.

Revere, Mass.

Low Pressure Turbine Experience

Several years ago while in charge of a power house in southern Maryland, I had a peculiar experience with a 750-kw. low-pressure turbine. The turbine was set to run at 7-lb. pressure, but had a connection for high pressure (150 lb.), which would open automatically in case of a shortage of low-pressure steam. We had considerable trouble with the turbine because it took high-pressure steam at all times, and carried small loads as well as heavy ones. This waste of steam made much difference in the coal bills and it became necessary to remedy the trouble, due to this increasing expense.

The turbine received its low-pressure steam from three engines exhausting into a receiver and these through a separator into the turbine. This separator was of the vertical type about 7 ft. high and 3 ft. in diameter, and as there was not room to spare, it was set down under the floor and extended into the ground. We took the turbine

apart and found all the blades in good condition, except that they were covered by a thin coating of magnesia, and it was thought that this may have made some difference in the steam consumption, but after running the turbine a few weeks we again experienced the same trouble.

Several weeks later a copper expansion joint between the separator and the turbine sprung a leak and as we had to tear the concrete floor up to get at it, I decided to examine the separator. After removing the top plate, we found that it contained a large screen made of sheet steel punched full of holes, and over each hole there was a piece of steel that acted as a shield. These holes were completely stopped up with what we took to be soft packing that had worked through from the engines. After thoroughly cleaning this out, which took about two days, we put the separator together again. We also lost no time in putting brass rings in back of each piston-rod stuffing-box to keep the packing where it belongs. This undoubtedly prevented the packing from being drawn into the separator as no trouble from plugged screens was again experienced, and the coal consumption immediately decreased.

PAUL B. PRUTZMAN.

North Wales, Penn.

Engineers' License Law Violation

There was a case in court in this district for violation of the engineers' license law. The defendants were represented by a lawyer, while the state inspector conducted the case for the Commonwealth.

It came out at the trial that the defendants were operating a first-class plant with a special license, which was issued for the operation of the old engine. In the meantime they had installed a steam turbine having a steam inlet of such size that the law called for an operator with a first-class engineers' license.

They claimed that they could not find a first-class man; that they had advertised in all the local papers without results. At this stage of the procedure the inspector said that he could and would furnish the defendants with a first-class engineer capable of taking charge of their plant.

After a whispered conference between defendants and lawyer, the lawyer asked:

"How much a week would such a man expect?"

Inspector: "About forty dollars."

Lawyer: "Now, Mr. Inspector, don't you consider $40 per week pretty steep for an engineer?"

Inspector: "No I don't. Do you consider yourself worth that amount per week?"

Lawyer: "Yes, but what has that got to do with this case?"

Inspector: "Only this, a man who gets a first-class engineers' license in this state has to study longer, and, in fact, know more than you do."

Lawyer: "Are you in the business of furnishing first-class engineers?"

Inspector: "No."

Lawyer: "Have you ever furnished a first-class engineer to any firm in this locality?"

Inspector: "Yes, one. This firm's superintendent claimed he could not get such a man, and I found one for him at his request."

Lawyer: "How much did this man get per week?"

Inspector: "Thirty dollars."

Lawyer: "Then why do you say that my clients would have to pay $40 per week?"

Inspector: "Because of the difference in the operation of the plants."

Lawyer: "Would you mind explaining what the difference is?"

Inspector: "No, I would not mind, but what is the use, you would not understand if I did."

JOHN ARMSTRONG.

Adams, Mass.

How Should New Valve Stems Be Put?

In the near future I must put in a new set of valve stems in our Corliss engine. As I have never before put in a set of valve stems I would like to receive full directions for doing this work from some reader who has had such experience. Will it be necessary to so disturb the adjustments to the valve gear as to require the valves to be reset after the new stems have been put in? Keyways must be cut in these new stems before they are put in.

H. W. LEE.

Lynchburg, Va.

[Ordinarily all that should be necessary is to cut the keyseats in the new valve stems in the same place that they are located in the old stems. The new stems could then be put in without disturbing the adjustment of the valve gear.

Perhaps some reader will comment.—EDITOR.]

Analogy for Initial Condensation

It is well known that one of the principal sources of loss in a steam engine is that due to initial condensation, or heat lost from the steam as it condenses on the comparatively cool cylinder walls on entering the engine. This loss or heat interchange between the steam and the cylinder can be greatly reduced by using superheated steam and to understand why this should be so, perhaps the following rather crude illustration will prove of assistance.

Imagine yourself equipped with a thousand ivory billiard balls at 100 deg. F., which you throw against a piece of armor plate at 60 deg. F. The perfectly elastic balls bound away so quickly that they give little of their heat to the plate, which remains nearly stationary in temperature. Now change a half of the balls for soft putty balls of the same size and temperature. Instantly a putty ball hits the iron it flattens out and sticks, giving up all its available heat to the iron surface.

The ivory balls represent molecules of superheated steam in such a stable condition that they are able to withstand impact against a colder surface without losing their shape or elasticity. The putty balls, on the other hand, correspond to steam molecules, which collapse or condense, on striking the iron to which they give their heat on condensation.

G. W. MUNRO.

LaFayette, Ind.

The 22,000-b.hp. Zoelly turbine, which is being built for the Manchester (England) Corporation power station, will have only 14 stages as compared with 20 stages in the previous 6000-kw. sets.

QUESTIONS BEFORE THE HOUSE

Piston Rod Area in Horsepower Calculations

In the reply to J. N., in the Nov. 12 issue (Inquiries of General Interest), relative to neglecting the area of the piston rod in calculating the horsepower of an engine, I note that you say that "for all practical purposes it may be neglected, but to be accurate it should always be considered."

Your reply recalls the method used by one firm in calculating the horsepower from the diagrams. It consists simply of subtracting one-half the area of the piston rod from the area of the piston. It will be seen that as on one side of the piston we have no piston rod and on the other we have the piston-rod area, this process balances the areas.

To take an example from practice: In a certain marine engine the high-pressure cylinder diameter is 31 in., stroke 5 ft. 6 in. and the diameter of the piston rod 6¼ in. On one run the mean effective pressure, as shown by the indicator diagrams, was 51.6 lb. and the revolutions per minute 50. Now neglecting the area of the piston rod we have

Area of cylinder = 754.77 sq.in.

Cylinder constant equals

$$\frac{2LA}{33{,}000} = \frac{2 \times 5.5 \times 754.77}{33{,}000} = 0.2516$$

$$i.hp. = (51.6 \times 0.2516) \times 50 = 649.1$$

Again considering the area of the piston rod we have, area of cylinder — ½ area of piston rod is

$$754.77 - 15.33 = 739.4 \ sq.in.$$

Cylinder constant

$$\frac{2 \times 5.5 \times 739.4}{33{,}000} = 0.2465$$

$$i.hp. = (51.6 \times 0.2465) \times 50 = 636 \ i.hp.$$

This shows that an error of 2.06 per cent. too much is made when the area of the rod is not deducted from the area of the crank side of the piston. This result, however, is only 0.7 per cent. lower than that given in your answer to J. N.

J. H. George Morrison.

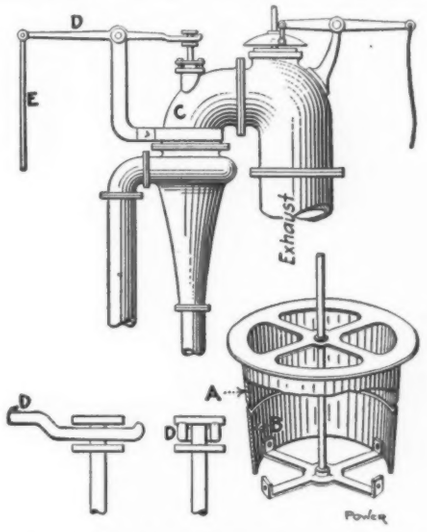
Glasgow, Scotland.

Improvement to Old Stvle Siphon Condenser

Relative to the article headed "Common Troubles of Siphon Condensers," page 615 of the Oct. 22 issue, the following description of an improvement that greatly facilitates cleaning the condenser cone during the operation of the condenser may be of interest.

The brass cone was removed from the condenser head and cut in the middle. The upper half *A*, was turned down in a lathe and a cut taken off the inside at the top of *B*. This allowed *B* to freely slip over the outside of *A*. A cast-steel spider was made to fit the bottom of *B* and was held by screws. A ¾-in. steel rod was passed through the center of the spider in *A*, as shown. By drilling a hole in the elbow *C*, and fitting a stuffing-box where the eye-bolt was located, the lower half of the cone could be raised while the pump was running and all obstructions washed down into the water-leg.

A lever *D* forked to take in the ¾-in. rod, that had collars above and below the fork, served to raise and



Siphon Condenser Cone Remodeled to Facilitate Quick Cleaning

lower *B*, and a piece of pipe *E* on the other end of the lever enabled one to work the lever from the floor.

This arrangement has worked for about a year without causing any trouble. The saving in labor and gaskets is well worth the trouble of altering the condenser to make the improvement.

E. B. Tifft.

Windsor Locks, Conn.

Removal of Scale from Turbines

In reply to Mr. Wood, who requested information in the Dec. 31 issue as to particulars of our way of removing scale from a turbine, I would advise as follows:

At the present time we are operating 65,000 kw. in units as follows: Three 8000-kw., one 12,000-kw., one 14,000-kw., and one 15,000-kw. unit; all are vertical turbines except one 8000-kw. reciprocating engine. Our boilers are all B. & W. water-tube type equipped with superheaters and fired by underfeed mechanical stokers using forced draft. The boilers have a rated normal capacity of 1000 kw. with two to three inches of water on the draft gage; they carry 180 lb. steam pressure with about 75 deg. F. superheat and the steam is delivered at 170 to 175 lb. with about 25 deg. F. superheat at the throttle. The condensers, with but one exception, are of the counter-current type equipped with displacement rotary pumps.

Our accumulator pressure is 1000 lb. per sq.in., which gives about 750 lb. at full load on the step bearings; this is allowed to approach within 100 to 150 lb. of the accumulator pressure before any cleaning off of scale on the turbine vanes is attempted, as we consider 100 lb. difference a reasonably safe limit.

The maximum quantity of coal oil used in this cleaning is six barrels for a 15,000-kw. turbine, the quantity varying with the size of the machine and the judgment of the engineer in charge as to the amount of scale present as indicated by the step pressure gage and also by the time elapsed since the last cleaning. While this cleaning is being done the turbine is allowed to exhaust straight to the atmosphere through the high-pressure relief valve at the condenser base, thus cutting out the condenser altogether. No appreciable heating of the turbine itself is experienced.

The feed water used is from the Schuylkill River, which, under normal conditions, gives the following analysis, subject to more or less change, due to high or low water, spring freshets, etc.:

PARTS PER MILLION	
Organic and volatile matter	30.3
Inorganic residue	186.3
Total solids	216.6

Further subdivided as follows:

PARTS PER MILLION	
Sodium chloride	19.86
Sodium sulphate	50.95
Calcium sulphate	36.57
Calcium carbonate	46.63
Magnesium carbonate	57.69
Silica, alumina, ferric oxide, potassium chloride, etc	4.90
	216.60

To make this a good boiler feed water, certain chemicals, such as soda ash, caustic soda, etc., must be added in moderate quantities. A certain amount of "Kutch" is also sometimes used to obtain the amount of tannic acid that will keep the scale from adhering to the tube.

I trust this information will be of benefit to Mr. Wood and others who are troubled with scale in turbines.

MORGAN G. JOHNS.

Philadelphia, Penn.

Design of Surface Condensers

I agree with the general principle of the design of surface condensers as advocated by Mr. Booth in the Oct. 22 issue, if this is not carried to extremes. This is undoubtedly the opinion of all other condenser designers.

Mr. Booth is apparently writing from an idealistic standpoint, but he wisely qualifies his statements to some extent by the following sentence: "Except for considerations of cost and space, there is no reason why a condenser should not be many times its ordinary cubical capacity with widely spaced tubes dispersed throughout the space."

The second of the reasons quoted would appear to be of greater importance, but I would suggest that there is at present no known reason why the cubical capacity should attain the dimensions suggested. In fact, the general tendency today is for surface and volumetric capacity to be reduced rather than increased as a result of more careful design, enabling much higher heat-transmission rates to be adopted.

Undoubtedly the cost has considerable bearing on the proportions of surface condensers, but the mere increase of volumetric capacity may be easily obtained by retaining what is practically the standard width or diameter, and increasing the length of the shell at relatively little additional cost. The major expense in a surface condenser is that due to the tubes and tube plates, and by increasing the length of the shell to obtain greater volumetric capacity only the additional length of tubes would have to be considered.

This arrangement would give an opportunity to increase the tube pitch as recommended by Mr. Booth, but there is undoubtedly a limit beyond which the pitch should not be increased.

One of the principal factors in efficient heat transmission and condensation is a lively molecular motion of the condensing medium and the steam. This fact is generally recognized and acted upon by most of the well known manufacturers of condensing plants.

The important point to bear in mind when proportioning condensers is not so much volumetric capacity as the provision of a perfectly free steam entry to the first and immediately adjacent rows of tubes. Consideration of this statement will show that not only must the tube pitch be carefully proportioned, but the ratio of the length of the shell to the diameter or width, as related to the exhaust-steam inlet branch, must be considered. The exhaust inlet should preferably be arranged so as practically to conform to the shape and dimensions of the condenser top. This is, however, fully recognized as an expensive and awkward arrangement when the space usually provided for condensing apparatus is considered; so the next thing to do is to arrange the exhaust branch on a dome of suitable area in the middle, gradually tapering away to the ends and sides of the shell, thus distributing the steam over the whole inlet surface without the aid of internal baffles.

Practically all manufacturers of standing carefully proportion the pitch of the tubes to provide sufficient steam area based on velocities determined by experiment and depending upon the quantity of steam flowing and the vacuum to be maintained. Steam baffles are now reduced to a minimum; in fact, one manufacturer at least provides only one baffle, and this is in the condenser base.

I have met with condensers wherefrom the removal of certain tubes has proved advantageous, but these have been of the old-fashioned marine box-of-tubes condensers, but I have yet to see a modern surface condenser under proper operating conditions having a difference in vacuum equal to 1 in. Hg. between the inlet and outlet.

The following particulars from actual condensers built by different firms indicate the lines on which they work:

(1) 29.1-in. vacuum; 30-in. barometer. Exhaust-steam inlet velocity 530 ft. per sec., steam velocity between first row of tubes, 250 ft. per sec. In this condenser three tube pitches are used, i.e., 1¼, 1⅛ and $1\frac{1}{16}$ in. for ¾-in. diameter tubes.

(2) 28-in. vacuum; 30-in. barometer. Exhaust-steam inlet velocity 260 ft. per sec., steam velocity between first row of tubes, 140 ft. per sec. In this condenser two tube pitches are used, i.e., 1⅛ and $1\frac{1}{16}$ in. for ¾-in. diameter tubes.

Both the above condensers are highly efficient in operation, the circulating water leaving the condenser at a temperature 3 to 4 deg. F. below that of saturated steam at the vacuum carried.

W. VINCENT TREEBY.

Goodmayes, Eng.

INQUIRIES OF GENERAL INTEREST

Polarity of Dynamo—In a dynamo, what determines whether a pole should be N or S, and how may the poles be reversed?

R. G. M.

The polarity is determined by the direction in which the field current flows around the magnet and the poles may be reversed by reversing the direction of the current.

Determining Flow of Water Through Pipe—What is a plain and easily understood formula for determining the amount of water which will flow through a pipe?

S. D. G.

There is no simple formula for figuring this flow. No formula can be absolutely accurate as the rough pipe surfaces are extremely variable, due to the different makes of new pipe, rough interior surfaces and reduction in interior surface diameter resulting from use. Many engineers prefer Darcy's formula, which is

$$h = \frac{(0.0198920 + 0.00166573)\frac{l}{d}}{a} \times \frac{v_s}{2g}$$

in which

h = The loss of head in feet due to friction.
d = The internal diameter of pipe,
v = The velocity in feet per second.
l = The length of pipe in feet.
2g = 64.324.

Illuminating Gas for Heating Purposes—What are the advantages and disadvantages of illuminating gas compared with coal for making steam to heat a small plant that requires 1000 sq.ft. of radiation?

W. T.

Gas burned under the boilers instead of coal would require very little attention after starting the fire. It is cleaner than coal, and if properly introduced into the furnace would produce no smoke. Besides, there would be no standby losses such as banking a coal fire, for the minute the service was no longer needed it could be shut off immediately and started again just as readily. The principal objection to illuminating gas for this purpose is its excessive cost when compared to coal. Ordinarily, a cubic foot of illuminating gas contains about 650 B.t.u. A good quality of coal will contain 14,000 B.t.u. per lb., so that in heat units a pound of coal is equivalent to

$$\frac{14,000}{650} = 21 \text{ cu.ft. of gas}$$

Gas, however, will burn at a little higher efficiency than coal and there will be no standby losses. It may therefore be assumed that the efficiency of combustion is 10 per cent. higher than that for coal. In reality, 19 cu.ft. of gas will be equivalent to 1 lb. of coal. Multiplying by 2000 will give 38,000 cu.ft. of gas as the equivalent of a ton of coal. A fair price for illuminating gas is 80c. per thousand cubic feet, so that 38,000 cu.ft. would cost approximately $30, as compared to $4 or $5 for a good grade of coal. The excessive cost of the gas would thus eliminate its use.

Waste in Excess Condensing Water—How much money is wasted in a year where such an excess of condensing water in the vacuum pump is required that the feedwater heater constantly overflows at a rate of about one gallon a minute of water that has been heated from 62 to 200 deg. F., if this waste lasts 18 hr. a day and six months in the year, and coal costs $2.90 a ton?

E. T. B.

As the total time is

$$182\tfrac{1}{2} \text{ days} \times 18 \text{ hr.} \times 60 \text{ min.} = 197,100 \text{ min.}$$

with the discharge rate of condensing water 1 gal. per min., a gallon of water being $8\frac{1}{3}$ lb., the total weight of heated water lost would be

$$197,100 \times 8\tfrac{1}{3} = 1,642,500 \text{ lb.}$$

Each pound of water being heated from 62 to 200 deg. F. would receive

$$200 - 62 = 138 \text{ heat units}$$

and the total heat lost would be

$$1,642,500 \times 138 = 226,665,000 \text{ B.t.u.}$$

From one pound of good coal and 75 per cent. boiler efficiency may be realized about 10,500 heat units. Therefore,

$$\frac{226,665,000}{10,500} = 21,587 \text{ lb.}$$

of coal would be required: that is,

$$\frac{21,587}{2000} = 10.79 \text{ tons}$$

of 2000 lb., which at $2.90 per ton would amount to $31.29.

Calculating Horsepower Without Indicator—What will be the maximum indicated horsepower developed by an engine with 20-in. cylinder, 36-in. stroke, receiving steam at 125 lb. pressure, cutting off at ¼ stroke, exhausting to the atmosphere and running at 90 r.p.m. Also when running at 110 r.p.m.?

F. P. L.

First find the mean effective pressure. This will be the difference between the mean forward pressure and the mean back pressure. The mean forward pressure can be found in Low's "Steam Engine Indicator," page 115, or by calculation from the formula

$$P_m = (1 + \log_e R)(f + c) - c$$

where

P_m = Mean forward pressure per pound of initial pressure
$\log_e R$ = Hyperbolic logarithm of the ratio of expansion;
f = Fraction of stroke complete at cut off;
c = Clearance per cent.

The clearance is not given, but may be assumed to be 5 per cent.; f was given as ¼ or 0.25.

The ratio of expansion would nominally be 1 divided by the fraction of the stroke complete at cut off or $\frac{1}{0.25}$ but corrected for clearance becomes

$$\frac{1 + 0.05}{0.25 + 0.05} = \frac{1.05}{0.3} = 3.5$$

From a table of hyperbolic logarithms $\log_e R$ is found to be 1.2528.

Now, substituting in the formula

$$P_m = (1 + 1.2528)(0.25 + 0.05) - 0.05$$
$$= (2.2528 \times 0.3) - 0.05 = 0.6258 \text{ lb.}$$

The total mean forward pressure will therefore be P_m multiplied by the initial pressure, which is given as 125 lb.

$$0.6258 \times .125 = 78.225 \text{ lb.}$$

From this must be subtracted the mean back pressure. As neither this nor the point of the beginning of compression is given, it may be assumed to be 2 lb. above the atmosphere, making the total back pressure

$$15 + 2 = 17 \text{ lb.}$$

Subtracting this from P_m gives

$$78.225 - 17 = 61.225 \text{ lb.}$$

as the mean effective pressure. This is the P. in the well known formula for horsepower

$$\text{Hp.} = \frac{\text{Plan}}{33,000}$$

where

l = Length of stroke in feet;
a = Area of piston in sq.in.;
n = Number of strokes.

l was given as 36 in. or 3 ft. a is equal to the area of a circle 20 in. in diameter or

$$a = 0.7854 \times 20^2 = 314.16 \text{ sq.in.}$$

n is equal to twice the revolutions per minute or

$$90 \times 2 = 180$$

Now, substituting in the last formula

$$\text{Hp.} = \frac{61.225 \times 3 \times 314.16 \times 180}{33,000} = 314.745$$

The horsepower at 110 r.p.m. may be obtained by dividing the above by 90 and multiplying by 110.

$$\frac{314.745 \times 110}{90} = 384.688 \text{ hp.}$$

ENGINEERS' STUDY COURSE

LOGARITHMS—PART III

In the two preceding lessons on logarithms we learned how to find the logs of given numbers and the numbers corresponding to given logs. In this lesson we will proceed with some practical examples for the working of which it will be necessary for the student to procure a table of logarithms. As mentioned before, these tables may be found in most engineers' handbooks.

RULE: *To multiply numbers, add their logarithms. The sum will be the log of the product.*

EXAMPLE: What is the circumference of a flywheel 24.5 ft. in diameter? To find the circumference we multiply the diameter by 3.1416.

The characteristic of 24.5 being 1 less than the number of integer places is 1, and that of 3.1416 is 0. The mantissa of each log is found from the table and the complete logs are as follows:

log of 24.5 = 1.3891661
log of 3.1416 = 0.4971509

1.8863170
0.8863158 = log of 76.969

12

Adding we find the log of the product to be 1.8863170. The nearest log to this in the table is 8863158, which corresponds with the number 76,969, and the characteristic being one we point off

$$1 + 1 = 2$$

places of whole numbers, making the number 76.969. Obviously there is no need of carrying this number out to a greater number of decimal places.

RULE: *To divide, subtract the log of the divisor from the log of the dividend. The remainder will be the log of the quotient.*

EXAMPLE: How many gallons in a tank containing 80,619 cu.in.?

One gallon contains 231 cu.in. Hence we must divide 80,619 by 231.

log of 80,619 = 4.9064374
log of 231 = 2.3636120

2.5428254 = log of 349

Here the characteristic of the log of the difference is 2. The mantissa is found in the table and corresponds to the number 34,900; and as the characteristic is 2 we point off three integer places in the number, and as the ciphers have no value we reject them.

Instead of subtracting we might have added the arithmetical complement of the log of 231, i.e., the difference between that log and zero

$$0 - 2.3636120 = -7.6363880 - 10$$

and then subtract 10 from the sum thus,

log of 80,619 = 4.9064374
ar. com. of log. of 231 = −7.6363880

12.5428254

12.5428254
10.0000000

2.5428254

This is the same result we got by subtracting. This method is particularly useful when multiplications and divisions occur in the same example.

What horsepower would be developed by an engine 24 in. in diameter, 46-in. stroke running at 75 r.p.m., with 38.27 lb. m.e.p.?

P	log of 38.27	=	1.5828585
L	log of 46	=	1.6627578
	ar. com. log of 12	=	−2.9208188
A	log of 24	=	1.3802112
	log of 24	=	1.3802112
	log of 0.7854	=	−1.8950909
N	log of 150	=	2.1760913
	ar. com. of log of 33,000	=	−5.4814861
			2.4795258
			177 = 301.66 hp.

The arithmetical complement may be written directly from the table by putting down instead of the figures there given, the difference between them and zero, which will be for the first or right-hand figure the difference between the number as given and 10, and for all the rest the difference between the given figure and 9. If the characteristic is plus, i.e., if the quantity dealt with is an integer the characteristic will be minus, and one more than the characteristic of the log, as seen below

0.0000000
log of 12 = 1.0791812
−2.9208188

0.0000000
log of 33,000 = 4.5185139
−5.4814861

When the characteristic of the log is minus, i.e., when a fraction is dealt with

0.0000000
log of 0.7854 = −1.8950909
0.1049091

the characteristic will be plus one less than the characteristic of the logarithm.

0.0000000
log of 0.00075 = −4.8750613
3.1249387

RULE: *To raise a number to any given power, multiply its log by the exponent of the power.*

If we wish to raise 25 to its third power we would have to perform the operation

$$25 \times 25 \times 25 = 15{,}625$$

This example could be done by multiplying the log of 25 by 3, thus,

log 25 1.397940
3
log 25^3 = 4.193820 = 15,625

Suppose we wanted to find 6^8, the eighth power of 6, we would have to do much figuring to get this by ordinary arithmetic. By the use of logarithms it is very easily obtained. We multiply the log of 6, which is 0.7781513 by 8, which gives 6.2252104, and the corresponding number is found to be 1,679,616.

Many problems are met with where it is necessary to raise a number to a fractional power. This operation is usually very difficult by ordinary arithmetic, but is easily performed by the use of logarithms. For example, you have often seen the expression $PV^{\frac{17}{16}} =$ a constant. This expression means that the value of P is to be multiplied by the value of V raised to the seventeenth power and the sixteenth root extracted.

We raise the value of V to the seventeenth power by multiplying its logarithm by 17, then dividing the product by 16, we get the logarithm of the sixteenth root of the seventeenth power. Having obtained this log, all that is necessary is to find its corresponding number.

Usually it simplifies matters to reduce fractional exponents to decimal form. Thus the above $\frac{17}{16} = 1.0625$, which is the exponent.

Suppose we want to find the value of $746^{\frac{7}{8}}$. We must raise 746 to the seventh power and extract the eighth root.

By logarithms it is done thus:

$$\text{The log of } 746 = 2.8727388$$
$$7$$
$$8)20.1091716$$
$$2.51364645$$

The number corresponding to the log 2.51364645 is 326.222 ans.

To extract the root of a number we divide the log of the number by the index of the root, and find the number to which this log corresponds.

Example: Find the cube root of 3.1416.

$$\text{log of } 3.1416 = 0.4971509$$
$$3)0.4971509$$
$$0.1657169$$

The number corresponding to log 0.1657169 = 1.46459.

Example: $\sqrt[6]{272} = \text{log of } 272 \div 6.$

$$\text{log of } 272 = 2.4345689$$
$$6)2.4345689$$
$$0.4057615$$

$$0.4057615 = \text{log of } 2.545432$$

In this last example we carried the number out to six decimal places, but for all ordinary calculation three places is sufficient.

Instead of writing a minus sign before the characteristic when it is negative, it is common to express it as

$$9.\ mantissa - 10$$

However, when the cube root is to be found, this involves a fraction when dividing by 3; therefore it is better to use for minus 1 ($\bar{1}$)

$$29.\ mantissa - 30$$

which is equivalent to characteristic 9 minus 10.

Example: Find the cube root of 0.124.

$$\text{log of } 0.124 = \bar{1}.093421$$

Instead of expressing this as 9.093421 — 10, we write it

$$29.093421 - 30$$

Now dividing by 3

$$3)29.093421 - 30$$
$$9.697807 - 10$$

Now we find the number corresponding to the mantissa

$$697807 = 49{,}866$$

As our characteristic is negative we know that this number is wholly decimal. Therefore,

$$0.49866 = \sqrt[3]{0.124}$$

The student can now appreciate the value of logarithms for raising numbers to high and fractional powers and for extracting cube root and all higher roots. Regarding the extracting of roots above cube root a good thing to remember is that logarithms are nearly always necessary.

In the Dec. 14 issue it was stated that when the index is an odd number, subtract 3 from it and divide the remainder by 2, extract the square root that number of times and then take the cube root. As an example it was stated.

"To get the seventh root

$$7 - 3 = 4;\ 4 \div 2 = 2$$

So we take the square root twice and then extract the cube root, and the sum of all these indices or exponents will be

$$2 + 2 + 3 = 7."$$

This was a mistake. The subtracting of 3 from the exponent gave the quotient of the original quantity divided by the cube of that quantity, not the cube root. The student should use logarithms in solving such problems.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

An "Engineering News" correspondent is dead set against the "reversed dam." He says "its stability depends upon its being perfectly glued to a solid rock foundation"; that he's "intensely serious about this matter." Sure! Must have been something wrong with the glue in all the reversed dams we've ever seen. What is this dam glue, anyhow?

Blink Boffum, the popular young electrician at the Bunktown lighting plant, takes his pen in hand to say that he has fully recovered from his "fall" off the H_2O shay. Blink denies the report that he was pushed off, says he just sort of slipped down. But he's clumb back, and swears he's agoin' to stay put.—"Bunktown Progressive."

The staff writers on the Jewish newspapers in New York City struck recently for a five-hour day, overtime, and a few other modest demands. How lucky we engineers are! We can work as long as we want to—and even longer. It's one wild bacchanalian revel with us. We ought to pity these poor scribblers who are thus rudely torn from their benches after five hours' light exercise and forced to idle about until tomorrow.

An Arizona workman was caught in a flywheel, whirled about, thrown into a pit 20 ft. away, and then got up uninjured and went back to work. Then he kicked because his jumper was torn! No gratitude in some people. He ought to be glad he wasn't docked for the time he lost while aimlessly flying around this way.

At the time of the economizer explosion at Saylesville, R. I., when the boiler plant was wrecked a team of horses attached to a wagon stood within 30 ft. of the building. Brick and other débris flew about the horses and half filled the wagon, and yet the horses stood their ground without moving. This must be an example of what people call horse-sense. Now, we suppose some brute will upset a really remarkable incident by informing us that the horses were deaf.

We are skeptical of the old saw which buzzes forth that "Music hath charms to soothe the savage breast." A German engineer, one Puritz, climbed into the Hamburg Opera House orchestra one night and slammed Musical Director Klemperer over the bean. It is just possible that the music lacked the necessary soothing syrupiness to control the tumult in Puritz' chest. The opera was "Die Gotterdämmerung," not a peaceful expression, to say the least.

Another slander! Newspaper says that "Limburger is the only cheese that has stuck to the common people, all the others being too high-toned." Exception, your honor! We are prepared to admit counsel's statement that the defendant is guilty of a high tone (See Exhibit "A"), but we take objection to plaintiff counsel that it (the cheese) has "stuck" to the common people. Any sensitive man will be quickly moved to tears as its first zephyr is wafted to his nostrils—but, stuck! Not unless he eats it. Objection sustained.

London dispatches make a great howdydo over King George having a chill. Godfreys crickets! anybody can have a chill who wants it—and many have one wished on them. If a king can't have a chill during the reign, what's the use of—Don't mind'em, friend king, the papers are always pestering some one or other. Come on down to the plant after you knock off for the day; we'll fix you up.

Independent Air Pump for Condenser*

SYNOPSIS—Its advantages and development and a comparison between rotary and direct-acting pumps.

The most obvious advantage of an independent air pump is its ability to obtain a vacuum in the condenser before the main engines are started, thereby enabling the engines to be manipulated with greater ease and certainty at times when immediate response is of importance. On this account, the first adoption of such pumps was on board war vessels, and later for small river steamers, while the advent of the marine steam turbine made the provision of such pumps a practical necessity.

The earliest designs of independent air pumps were those in which the pump was driven by an independent crankshaft engine, and when the variable and peculiar nature of an air-pump load is considered, it will be obvious that a meager degree of success was possible, and that many failures resulted. Greater success was obtained when the air pump was coupled to the independent circulating engine, as these engines were under a considerable continuous load, and the variations of the air-pump load were not so vital in their effect, but even this arrangement was frequently unsuccessful, particularly where lack of care was shown in the details of design.

The direct-acting independent air pump was originally developed in the United States, while in its details it was perfected in Great Britain and in the form of the "Twin" air pump thoroughly satisfactory results were obtained. A further degree of simplification in design was the advent of the "Monotype" pump, in which a single air barrel of the three-valve type was driven by a single steam cylinder, and even now this constitutes a very successful type of pump for certain conditions. The suction-valveless pump, the features of which were originally enunciated in 1879, and which is now known as the Edwards air pump, offers distinct advantages as a type of air end for use as a lever-driven pump, but due to the principle of working, it is unsuitable for independent drive, and accordingly the old three-valve air pump has still maintained its position as the most efficient air end for independent working.

When, however, the principle under which air and water should be withdrawn from a condenser is considered, it is obvious that the "Twin" or "Monotype" pumps require to be very large on account of their handling the air and water together. Where of small dimensions, an undue degree of cooling of the feed water is necessary to enable them to obtain the desired vacuum, except under conditions of very slight air leakage. As already pointed out, it was early evident that the maximum efficiency would only be obtained by a combination of wet and dry pumps.

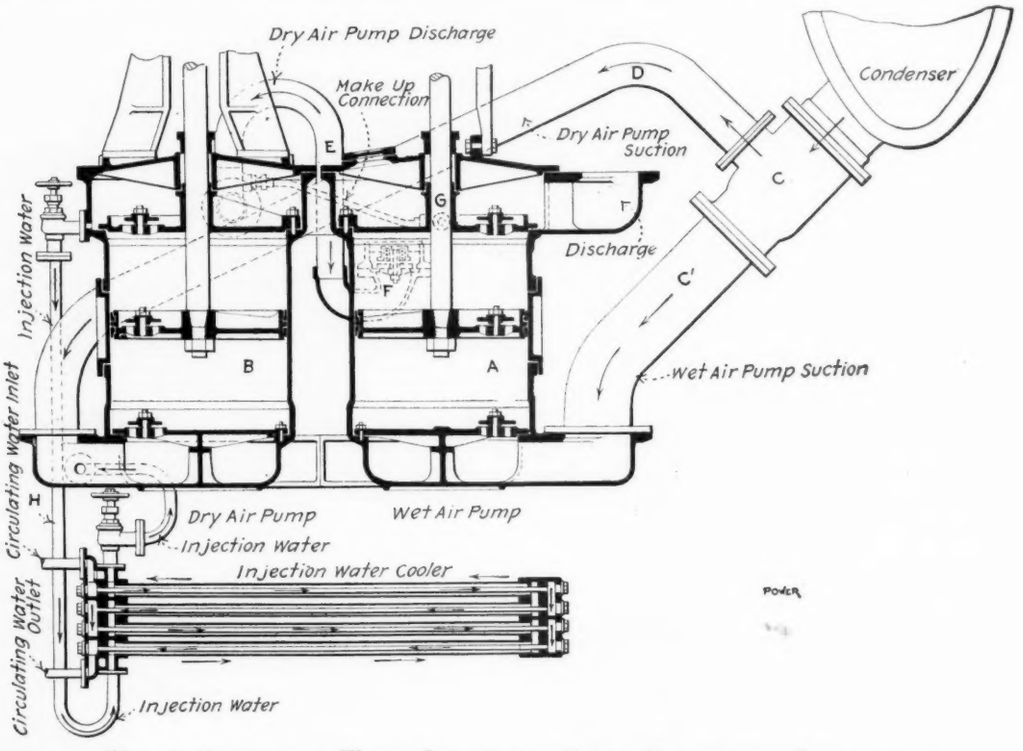


FIG. 1. SECTION OF WEIR "DUAL" AIR PUMP, COOLER AND CONNECTIONS

In the early stages of turbine propulsion many forms and combinations of wet- and dry-air pumps were fitted, but unfortunately these installations, in the majority of cases, were rendered apparently noneffective on account of faulty condenser design, and the present design of "Uniflux" condenser is a direct outcome of the apparent nonsuccess of the independent dry-air pump. Having succeeded in finding a satisfactory condenser design, the field was consequently clear for a fresh application of the wet- and dry-air pump principle. The main object was to secure a wet and dry pump, in one unit, in order to obtain an apparatus of minimum weight, of greater simplicity

*From paper read by William Weir before the Institution of Engineers and Shipbuilders in Scotland.

and steam economy, together with greater reliability than had been possible with the separate wet and dry pumps. These advantages have been secured in the "Dual" air pump in a simple and compact form.

Fig. 1 shows, in diagrammatic form, the arrangement of surface condenser, "Dual" air pump, and injection-water cooler. In all cases the wet pump *A* is situated below the steam cylinder, as this pump is the one which works under the maximum load. The dry pump *B* is driven by the beam and links in the usual way. One connection *C* is made to the condenser, but a branch pipe *D* is led to the dry pump, the connection being made in such a manner that the water will all pass by *C* to the wet pump. Apart from the separate suction arrangements to each pump, the "Dual" pump differs from the ordinary twin pump in that the dry pump discharges through the return pipe *E*, through a spring-loaded valve *F*, into the wet pump, at a point below its head valves. This spring-loaded valve is adjusted to maintain about 8 in. of mercury difference of pressure between the condenser, and what might be termed the hotwell of the dry pump.

The next point concerns the supply of water to the dry pump for water-sealing, clearance-filling, cooling and vapor condensation. When starting the pump, the filling valve *G* is opened for a short period, to enable the vacuum to draw in a supply from the hotwell of the wet pump. The valve is then closed and the water passes from the hotwell of the dry pump by the pipe *H* to an annular cooler through which a supply of sea water is circulated, and after being cooled the injection water passes into the suction of the dry pump, through the pump, and returns to the cooler in a continuous closed circuit; any excess caused by condensation passing over by the pipe *E* to the wet pump.

The advantages secured by such a combination are: A wet-air pump working approximately at the temperature due to the vacuum, in combination with a dry-air pump working at a much lower temperature, on account of which it is enabled to handle air leakage without any substantial cooling of the main body of feed water; a dry-air pump of high efficiency, due to it only requiring to discharge against 4-lb. pressure instead of 15-lb., as in the case of an ordinary air pump.

COMPARATIVE CAPACITIES AND WEIGHTS OF AIR PUMPS

Vessel	Condensed steam per hr. per pump	Rate in lb. per Hp. per hour	Type of Air Pump	Designed Vacuum inches	Volume swept by bucket Volume feed water	Lb. weight of pump per Hp. of main engine
S.S. "Kronprinz Wilhelm"	255,000	17	"Twin"	26	12.4	2.09
S.S. "Franconia"	90,000	15	"Dual"	26	7.4	1.102
Battleship (1902)	67,500	16	"Twin"	26	15.4	2.16
Battleship (1912)	94,500	14	"Dual"	28.5	13.3	1.377
Turbine Atlantic Liner	196,000	14	"Dual"	28.5	13.2	1.232
Destroyer (1907)	124,000	16	"Monotype"	26	12.7	0.765
Destroyer (1912)	177,625	14.5	"Dual"	28	8.8	0.466
Small Turbo-Generator Set	6,000	..	"Monotype"	28	33	....
Dredger Installation	15,000	..	"Monotype"	25	30	....
Three-Throw Edwards type for Electric Power Stations	.....	..	....	28.5	45.5	....
Weir "Dual" Air Pump for Electric Power Station	80,000	..	....	28.5	20.5	....
S.S. "Campania"	40,000	....	Engine-driven ordinary	25	60	....
Cargo Steamer	255,000	..	Engine-driven ordinary	25	45	....
S.S. "Rangatira"	40,000	..	Engine-driven "Dual"	25	26.5	....

The factors involved in the estimation of the correct size of air pump to employ, comprise in the first place considerations of the normal air leakage which the pump must deal with. This question is associated with the actual size of the installation. In a small installation the air leakage is always relatively higher than in a large one. It is further complicated by the nature of the installation.

The table shows the wide variation of practice, giving the capacity in terms of air-pump bucket volume swept to feed-water volume, also the weight of a number of air-pump installations, from which it may be noticed that the present-day degrees of high vacua are obtained with less weight and capacity than the older arrangements of, say, ten years ago. This arises in a certain degree from improved air-pump design and also to the greater care taken to avoid air leakage.

It is of interest to note the large difference in capacity between the ordinary engine-driven air pump and the independent air pump, while the capacity usually provided for electric-power-station practice appears excessive. The average steam consumption of independent "Dual" air pumps for vacuums of 26 to 28.5 in., expressed as a percentage of the feed water they handle, varies from 0.65 to 1.

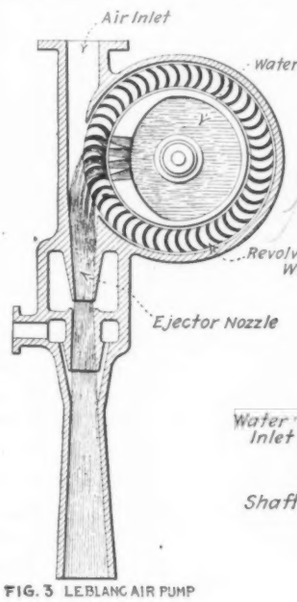


FIG. 3 LEBLANC AIR PUMP

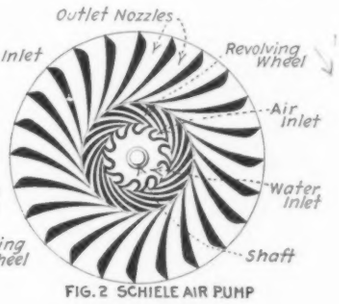


FIG. 2 SCHIELE AIR PUMP

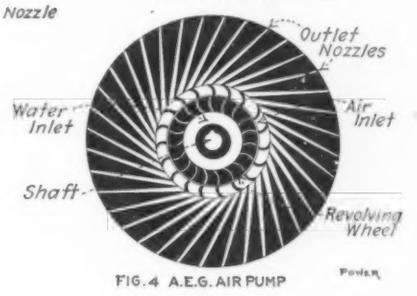


FIG. 4 A.E.G. AIR PUMP

During the last few years much interest has been taken in the development of rotary dry-air pumps, especially for electrical-station practice on the Continent; the two most notable examples are associated with M. Leblanc, of Paris, working in conjunction with the Westinghouse Co., and the rotary air pump constructed by the A. E. G. Co., of Berlin. While the utmost credit should be accorded to M. Leblanc for having developed the modern type of rotary air pump, it is of interest to note that in 1862, Christian Schiele, of Oldham, invented and built a rotary air pump regarding which he claimed:

"The system or mode of expelling air from condensers by combining or entraining it with injection water on its passage through a fan or centrifugal pump."

Fig. 2 shows the original Schiele pump, in which it will be seen that the water enters through a series of fixed openings, passes into a turbine wheel along with the air and vapor, in which it is given the necessary velocity to discharge it through guided passages to the atmosphere.

Fig. 3 shows a section of a modern Leblanc air pump in which a single ejector and nozzle is used for discharg-

ing the air and injection water, the combination or admixture of the two taking place outside the turbine wheel.

The A. E. G. design of rotary air pump shown diagrammatically in Fig. 4 is practically a Leblanc pump having a series of specially-shaped nozzles arranged circumferentially, the point where the air meets the entraining water being, as in the Leblanc pump, between the rotating wheel and the nozzles.

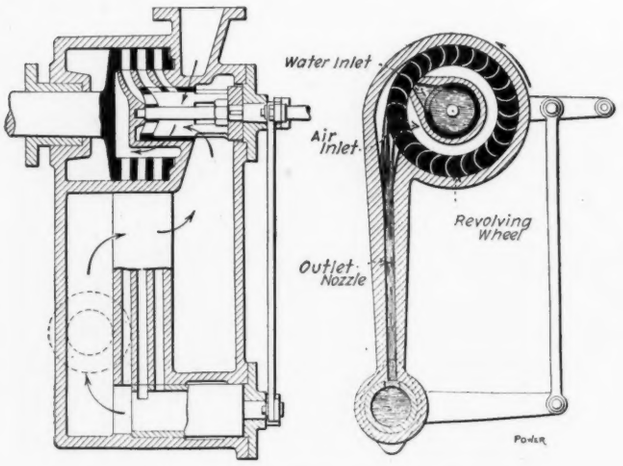


FIG. 5. WEIR ROTARY AIR PUMP

For marine use the adoption of rotary dry-air pumps has been comparatively slow, due in a large measure to the very onerous and varied requirements to which a marine pump has to conform, and further, to the extremely efficient nature of the existing direct-acting machinery. The necessity for driving such rotary machines by small steam turbines, in themselves still more or less in the development stage, necessarily implies high steam consumption, while even the best rotary air pumps require the absorption of a considerable power per unit of air removed. Further, the existing types do not appear to possess any degree of flexibility, in the sense that being velocity machines they must always run at the designed speed and accordingly use the same amount of steam, thus causing their performances at low powers of the vessel to be fairly inefficient.

Fig. 5 shows a recent development of the ordinary Schiele pump, which has been designed with the object of imparting to it a certain degree of flexibility. As will be seen, it differs from the Leblanc and A. E. G. types in that the air is brought into contact with the injection water in the wheel itself—somewhat similar to the Schiele pump—both passing through the wheel before being ejected through a simple nozzle; while in addition the wheel is constituted of four units, each being provided with a nozzle, and provision being made for closing any unit as desired, it enables the pump to run with whatever number of units are required to maintain the desired vacuum.

While it is probable that the entrainment type of rotary air pump may be further developed, its ability to handle heavy air leakage only by the use of considerable power rather handicaps it for marine work at present. Fig. 6 gives a representation of the characteristic performance of a modern marine rotary air pump at varying air leakage, while on the same diagram there is shown the performance of a direct-acting pump. As regards the proportions of the two pumps, their weights are identical, provided no allowance is made for the cooler and tank necessary for the rotary air pump. The relative steam

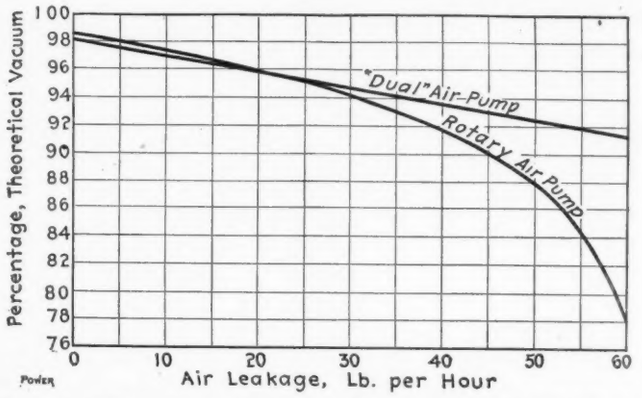


FIG. 6. "DUAL" AND ROTARY AIR-PUMP PERFORMANCES

consumptions of the two pumps, when doing the duty shown on the diagram, is as follows:

Direct-acting pump..... 800 lb. per hr.
Turbine-driven rotary pump... 3500 lb. per hr.

As may be seen from the diagram, the rotary pump maintains a very slightly higher vacuum than the direct-acting pump at small leakages, but rapidly falls off at large leakages.

[That portion of the paper devoted to feed-water heating and feed pumps will be published in a later issue.—EDITOR.]

Economizer Explosion at Saylesville

By the explosion of an economizer at the Glenlyon Dye Works, a branch of the Sayles Bleachery, Saylesville, R. I., at 4 p.m., Tuesday, Jan. 14, two men were killed and seven others were more or less seriously injured, one probably fatally. One section of the boiler house and the engine room were demolished.

A brick wall divided the power plant into two boiler rooms. In one, six 325-hp. water-tube boilers were installed, each connecting with a main steam header running the length of the boiler room. In the second boiler room there were four 72-in. return-tubular boilers. A second brick wall, at right angles to the first, separated this boiler room from an engine room in which there was a Corliss engine, directly connected to a generator. The 72-in. boilers were each connected by a 6-in. branch pipe to a 12-in. header. A smoke flue extended to the brick chimney, entering at the opening shown in Fig. 1.

Two economizers, side by side on an I-beam platform, supported by iron columns, were next to the partition wall. Beneath the economizers were the heater and boiler-room auxiliaries. The smoke flues of the water-tube boilers were connected to the economizers after passing through the brick wall between the two boiler rooms. As the economizers were in the boiler room containing the return-tubular boilers, it was here that the greatest damage was done.

At the time of the explosion only one of the two economizers was in service, the outside one being cut out for some slight repairs. When the working economizer exploded it demolished the second.

So severe was the explosion that the entire end of the power house, which was about 125 ft. square, and 30 ft. high was demolished, the roof collapsing in a tangled mass over the small boilers and débris. Fragments of the economizers were thrown about the yard. Part of the chemical laboratory at the rear of the boiler room was also wrecked.

As both economizers were blown to pieces, the one in service must have contained a large amount of water at a high temperature. Although the economizers were nearest the water-tube boilers, the dividing wall protected the latter to such a degree they were uninjured, and can be put into service at any time.

As the force of the explosion was directed toward the return-tubular boilers, that part of the power plant suffered most. The boiler fronts were demolished, and the steam-pipe connections were broken, as shown in Fig. 2. Happily the sudden release of steam from these boilers did not cause a second explosion.

Fig. 1 gives an idea of the damage done. In the foreground are parts of the economizers, smoke flue and general débris. The partition wall between the two boiler rooms is shown back of the 150-ft. chimney; a part of it was blown out. The chimney was only slightly damaged. The water-tube boilers are shown at the right of the chimney. The economizers stood on the platform just back of the chimney, Figs. 1, 2 and 4. Some of the larger sections of the economizer are shown in Figs. 2 and 3. No particular section appears to have been weaker than another; the headers were badly broken, the tubes broke off in the header in some instances, just outside in others, and at varying lengths, as shown.

Boiler-feed water was pumped from an open heater,

FIG. 1. GENERAL VIEW OF THE WRECKED BOILER HOUSE, SHOWING DAMAGE DONE BY THE EXPLOSION OF AN ECONOMIZER

FIG. 2. FIRING FLOOR IN FRONT OF RETURN-TUBULAR BOILERS

FIG. 3. PARTS OF THE ECONOMIZERS

Fig. 4, through the economizers; therefore the latter would be under boiler pressure. The cause of the accident is unknown. A number of experts have examined the ruins, but can give no authentic information. A representative of the economizer manufacturer holds to the theory that gases leaked into the idle economizer, and exploded with sufficient force to rupture the working unit, the explosion of which did the damage. Both were connected to a single smoke flue connected with the chimney.

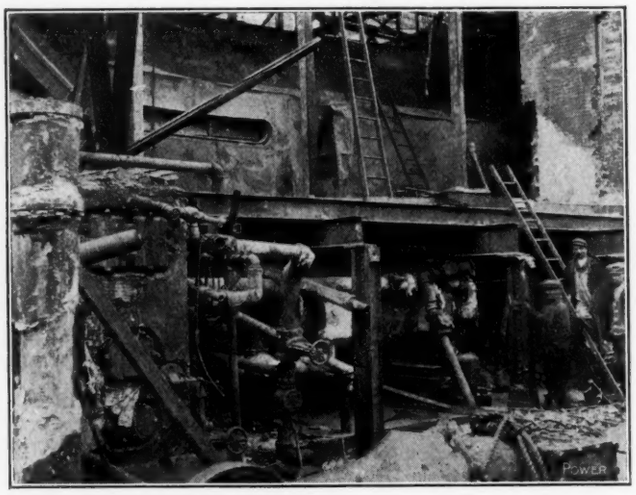


FIG. 4. AUXILIARY APPARATUS UNDER THE ECONOMIZER PLATFORM

So suddenly did the explosion occur that the two men who were killer were buried under fallen bricks and general débris where they stood. The body of fireman Hassan Momoth was not recovered from the ruins until 16 hr. after the accident, although search for the body was continued throughout the night. His body was finally located in front of the return-tubular boilers. August Blank, a piper, the other man killed, was found pinned to the concrete floor of the boiler room by a piece of roof girder. Death in both cases was doubtless instantaneous. Three men were released from the wreckage in about 20 min., a fourth, an ash man, was found in about an hour, unconscious and in a critical condition. Of the ten men in the power plant at the time of the accident, but one escaped uninjured. He was standing in the center of the boiler room near the heater when the explosion occurred.

Both Green economizers were 12 years old. The boilers and economizers were operated as they had been all day. They were insured by the Hartford Steam Boiler Inspection & Insurance Co. It is not known just what the loss will be, but it will amount to several thousands of dollars. It is doubtful if the return-tubular boiler will again be put into operation.

❖

Frank B. Monaghan's Memory Honored

In the presence of nearly 200 friends and associates, including many prominent labor officials, a monument was dedicated on Jan. 12, at Calvary Cemetery, Boston, Mass., to the memory of Frank B. Monaghan, organizer and a former president of the International Union of Steam Engineers. Inclement weather necessitated an early adjournment to the Revere House, Boston, where the exercises were continued under shelter. James Duncan, first vice-president of the American Federation of Labor and a close friend of Mr. Monaghan, delivered the dedication oration, and in the course of his address said:

Monuments throughout the world have generally been erected in the memory of military or naval heroes, and have been connected with the spilling of blood. This shaft has been raised for a nobler purpose, in the memory of one who devoted his whole being to the conserving of life and the betterment of the condition of humanity.

The monument is 9 ft. in height, sarcophagus in form, with a representation of a steam gage on its face, the indicator hand pointing to 50, the age of the deceased leader of engineers.

❖

International Refrigeration Exposition

It has been previously announced that during the third International Congress of Refrigeration at Chicago in September, 1913, an exposition of refrigeration will be held. The arrangements are perfected and the association thoroughly organized for holding this exposition in the amphitheater at Forty-second and Halsted Sts., the largest building of its kind in Chicago. It is equipped with trackage connection with all of the railways entering the city. Everything pertaining to refrigeration will be exhibited, cars, machinery of all sorts in active operation, fruits, game, poultry, meats, vegetables, eggs, butter and cheese. With the 500 or 600 delegates from Europe and South American countries and kindred conventions of American ice makers, meat packers, poultry, game, butter and egg people and others, nothing equaling the importance of this exposition has been undertaken since the World's Fair, says the committee:

The officers and directors are: President pro tem., William L. Wagner; vice-president, Frank L. Nickerson; general manager, William E. Skinner, with office at 819 Exchange Ave. Directors, including the officers: Theodore O. Vilter, John S. Field, F. S. Pilsbry, Arthur Meeker, H. C. Gardner, C. M. Secrist, George E. Haskell, A. D. White and H. W. Hart.

SOCIETY NOTES

The National District Heating Association will hold its fifth annual convention in Indianapolis, Ind., on May 27-29. The association promises a program of unusual benefit and interest, and urges a large attendance.

A. R. Maujer, Western editor of "Power," and a member of Illinois No. 1, lectured before the Chicago Association of Stationary Engineers on Wednesday evening, Jan. 15, on "Proximate Coal Analysis and Its Value in Power Plant Economy." The lecture was demonstrated with actual apparatus and the method employed clearly illustrated. This lecture will be entered in the national prize contest for the indicator outfit offered by the National Educational Committee.

PERSONALS

Manager E. Holcomb, of the Consumers Power Co., St. Paul, Minn., has been elected president of the Minnesota Electrical Association for 1913.

Staege & Bennett, consulting civil, steam and electrical engineers, Watertown, N. Y., have moved their offices from the Smith Building to the Charlebois Building, in that city.

Charles W. Naylor lectured before the Armour Institute Branch, N. A. S. E., on Jan. 15, taking as his subject the Marshall Field power plant, of which he is the chief engineer.

Herbert E. Stone is the New England representative of the Pittsburgh Piping & Equipment Co., piping contractors, engineers and machinists, with offices in the Equitable Building, Boston, Mass.

Errata: In the description of the Worthington Turbo Blowers and Compressors in the Jan. 7 issue, page 12, a very obvious mistake occurred. A turbo compressor referred to was described as operating against a pressure of 1000 lb.; it should have been 100 lb.